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Recent developments in ejector refrigeration technologies

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ABSTRACT

This paper aims at providing a literature review on the recent development in ejectors, applications of ejector refrigeration systems and system performance enhancement. The paper presents useful guidelines regarding background and operating principles of ejector. A number of studies are reported and categorized in several topics including, refrigerant selections, mathematical modelling and numerical simulation of ejector system, geometric optimizations, operating conditions optimizations and combinations with other refrigeration systems. Most of the works that have been carried out recently are still limited to computer modelling, more experimental and large-scale work are needed in order to provide better understanding for the real industrial application.

Contents

Abbreviations: AC, air conditioning; CFD, computational fluid dynamics; CHP, combined heat and power; CMRC, constant momentum rate change; COP, coefficient of performance; ERS, ejector refrigeration system; GCHP, ground coupled heat pump; GWP, global warming potential; HRVG, heat recovery vapour generator; MFG, multi-function generator; NXP, nozzle exit position; ODP, ozone depletion potential; PE, pressure exchange; TERS, transcritical ejector refrigeration system; TRCC, transcritical CO_2 refrigeration cycle

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1. Introduction

Refrigeration is recognized as an indispensable method of improving human beings' living conditions since early twentieth century. Refrigeration systems, in the various applications including food storage and provision of thermal comfort, have contributed significantly to the industrial and health sectors. Conventional vapour compression refrigeration cycles are driven by electricity with the consumption of fossil fuels. However, this results in air pollution and emission of greenhouse gases, and consequently poses a threat to the environment. Hence, improvement on the refrigeration system's working performance will result in less combustion of primary energy, and mitigation of the environmental pollution.

Ejector refrigeration systems (ERS) are more attractive compared with traditional vapour compression refrigeration systems, with the advantage of simplicity in construction, installation and maintenance. Moreover, in an ERS, compression can be achieved without consuming mechanical energy directly. Furthermore, the utilization of low-grade thermal energy (such as solar energy and industrial waste heat) in the system can helps to mitigate the problems related to the environment, particularly by reduction of CO_2 emission from the combustion of fossil fuels.

However, due to their relatively low coefficient of performance (COP) [1–3], ERS are still less dominant in the market place compared with conventional refrigeration systems. Therefore, in order to promote the use of ERS, many researchers have been engaged in enhancing the performance of ejector system and combining ERS with other refrigeration systems in order to improve the overall system performance. Building on other published review papers [1–3], this paper aims to update the research progress and development in ejector technology in the last decade. This paper will emphasize on the various combination of ejectors and other cycles. Linkages and comparisons between different research cases are presented, and similar study concepts are grouped and briefly described as overall summaries.

2. Performance characterization of ejector refrigeration system

A schematic view of a typical supersonic ejector is shown in Fig. 1. Normally, a steam ejector consists of four principal parts, the primary nozzle, the mixing chamber, the ejector throat and the subsonic diffuser. High pressure primary fluid (P) expands and accelerates through the primary nozzle, it fans out with supersonic speed to create a very low pressure region at the nozzle exit plane and hence in the mixing chamber. The high-velocity primary stream draws and entrains the secondary fluid (S) in to the mixing chamber. The combined streams are assumed to be

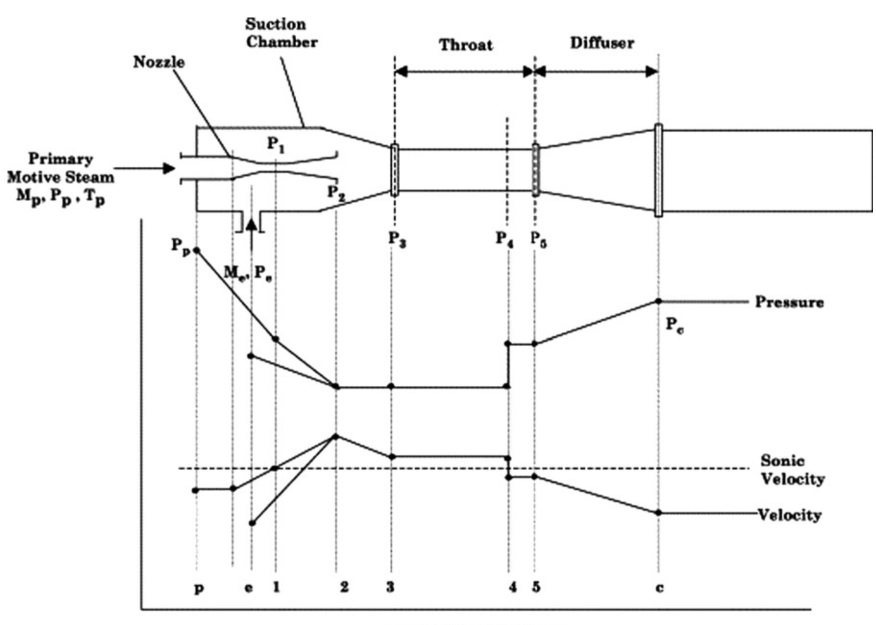

Fig. 1. Schematic diagram of a typical supersonic ejector [4].

completely mixed at the end of the mixing chamber and the flow speed is supersonic. A normal shock wave is then produced within the constant-area section, creating a compression effect, and the flow speed is reduced to subsonic value. Further compression of the fluid is achieved as the combined streams flow through the subsonic diffuser section.

There are several parameters used to describe the performance of an ejector. The working performance of refrigeration system is often measured by the coefficient of performance (COP), defined as the ratio between the cooling capacity at the evaporator and energy input at the boiler and pump:

$$\mathrm{COP} = \frac{\text{Cooling effect at the evaporator}}{\text{Energy input at the boiler and pump}} = \frac{Q_e}{Q_g + W_p}$$

where Q_e and Q_g are the cooling capacity at the evaporator and energy input to the generator, and W_P is work consumed by the mechanical pump.

As the energy input at the pump is usually negligible (typically less than 1% of the generator heat input), it is often not taken into account, and the COP is then calculated as the ratio of the cooling capacity at the evaporator and the heat input at the generator:

$$\mathrm{COP} = \frac{Q_e}{Q_g}$$

For refrigeration applications, the most important parameters are defined in terms of entrainment, expansion and compression ratios. The entrainment ratio (λ) is the ratio of secondary flow to primary flow. The expansion ratio is defined as the ratio of primary pressure to secondary pressure. The compression ratio gives the ratio of the compressed pressure to the secondary pressure. The entrainment ratio is related to the energy efficiency of a refrigeration cycle and the pressure ratio limits the temperature at which heat can be rejected. For a given primary pressure and secondary pressure, the highest entrainment ratio will be achieved when compression ratio is minimised.

2.1. Operating conditions

The operating conditions at various points (boiler, condenser and evaporator) are important parameters which affect the ejector's working performance.

The experimental studies carried out on the ejectors refrigerator [5] showed that, at each setting of boiler and evaporator operating condition, the operation of the ejector can be categorized into three regions (as shown in Fig. 2): double choked flow in the mixing chamber, primary chocking flow and reversed flow. When the ejector is operated under the "critical condenser pressure", COP and cooling capacity remain constant. Further increase in condenser pressure above the critical condenser pressure moves the thermodynamic shock wave into the mixing chamber and prevents secondary flow from reaching sonic velocity. Upstream conditions can now be transmitted downstream, which results in a reduction in secondary flow, entrainment ratio and COP. Eventually, secondary flow drops to zero, the ejector loses its function and primary flow will reverse back into the evaporator.

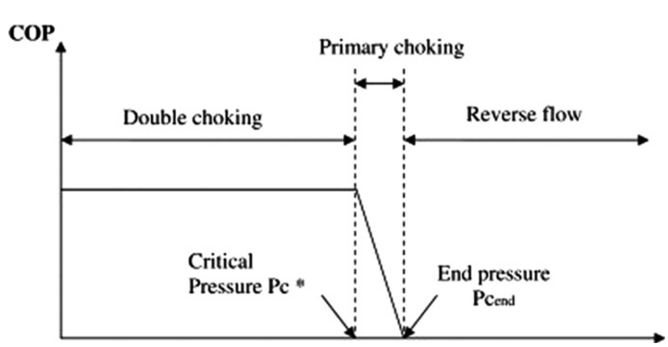

Fig. 2. Operational modes of an ejector [6].

In order to analyze the pressure change along ejector profile, Chunnanond and Aphornratana [7] carried out an experimental investigation of a steam ejector refrigerator. He concluded that there were two parameters which dominated the performance of an ejector refrigerator, the amount of secondary fluid passing through the mixing chamber and the momentum of the mixed stream. A decrease in the boiler pressure caused primary flow to decrease, resulting in an increase in the secondary flow. This caused the cooling capacity and COP to rise. However, this caused the momentum of the mixed flow to decrease. Thus, the critical condenser pressure was reduced. On the other hand, an increase in evaporator pressure, which was the ejector's upstream pressure, increased the critical condenser pressure. This also increased the mass flow through the mixing chamber and consequently increasing cooling capacity and COP. However, this would sacrifice the desired cooling temperature.

Selvaraju and Mani [8] carried out an experimental investigation on a R134a ERS. The results showed that with an increase in boiler temperature, entrainment ratio, COP and cooling capacity first increased and then decreased. Therefore, for the given condenser and evaporator temperatures, every ejector with particular configuration has an optimum boiler temperature, at which the maximum COP could be obtained. The same conclusion was presented by Ma et al. [6]. The author further stated that increasing the boiler temperature did not always increase system efficiency. An experimental investigation was carried out on a novel 5 kW ejector refrigerator suitable for solar energy applications. A spindle was introduced in order to control primary flow and provide a fine tuning of the ejector system. A maximum cooling capacity was found at a boiler temperature of 92.8 °C, while maximum COP was found at a boiler temperature around 90 °C.

Effects of non-dimensional parameters like compression ratio (pressure ratio of condenser to evaporator) and expansion ratio (pressure ratio of generator to evaporator) on the system performance were also studied by Sankarlal and Mani [9]. The results showed that COP increased with increase in expansion ratio and decreased with compression ratio.

An experimental study on a two-phase ejector in a refrigeration system was carried out by Chaiwongsa and Wongwises [10]. The effects of the external parameters i.e. heat sink (where condenser rejects heat to the cold water) and heat source (where evaporator is supplied by using hot water) temperatures and various relevant parameters on COP were also presented: cooling capacity varied inversely with the heat sink temperature whilst it varied proportionally with the heat source temperature.

According to Chen et al. [11], small droplets formed at the exit of the ejector could block the effective area and collide with the wall, causing damage. External superheating of the stream prior to entering the nozzle was believed to eliminate this effect. Compared with external influences, internal superheating caused by friction and normal shock wave affected system performance differently. Aidoun et al. [12] concluded that some degree of inlet superheat (around 5 °C) was necessary to prevent internal condensation but excess superheat was detrimental to the condenser efficiency at the exit.

The summarized operating conditions for recent simulation investigations are shown in Table 1.

2.2. Working fluids

The choice of the appropriate working fluid plays an indispensable role in the design of the ERS. The following requirements

Table 1
Experimental results of ejector refrigeration system under different operating conditions.

Reference	Cooling capacity (kW)	Working fluid	Evaporator temperature (°C)	Condenser temperature (°C)/pressure	Boiler temperature (°C)	COP	Conclusion
Chunnanond and Aphornratana [7]	3	Water	5:15	22:36	110:150	0.28:0.48	(1) A decrease in the boiler pressure caused the cooling capacity and COP to rise and the critical condenser pressure was reduced. (2) An increase in the evaporator pressure increased the critical condenser pressure, cooling capacity and COP, which sacrificed the desired cooling temperature.
Selvaraju and Mani [8]	0.5	R134a	2:13	26:38	65:90	0.03:0.16	(1) COP critical $= -0.375976R_{\mathbf{d}} - 0.284386R_{\mathbf{c}} + 0.242682\Phi + 0.933787$ ($R_{\mathbf{d}}$=driving pressure ratio, $R_{\mathbf{c}}$=compression ratio, Φ=ejector area ratio) (2) For given condenser and evaporator temperatures, every ejector with particular configuration has an optimum boiler temperature, at which the maximum COP can be obtained.
Sankarlal and Mani [9]	2	Ammonia	5:15	30:36	62:72	0.12:0.29	COP increased with increase in expansion ratio and decreased in compression ratio
Chaiwongsa and Wongwises [10]	1.8–3	R134a	8:16	26.5:38.5	50:60	0.3:0.48	Cooling capacity varied inversely with the heat sink (where condenser rejects heat to the cold water) temperature while it varied identically with the heat source (where evaporator is supplied by using hot water) temperature
Yapıcı et al. [13]	2	R123	0:14	108 kpa:142 kpa	83:103	0.12:0.39	(1) Optimum nozzle exit position was 5 mm outwards from the mixing chamber (2) Generator temperature higher than 97 °C resulted in constant cooling capacity but lower COP.
Ma et al. [6]	5	Water	6:13	25 mbar:65 mbar	84:96	0.17:0.32	(1) Increase the boiler temperature does not always accompany by increase in system efficiency. Maximum cooling capacity was found at boiler temperature around 90 °C. (2) Spindle can help to control ejector's primary flow and achieve fine tuning for system operation.

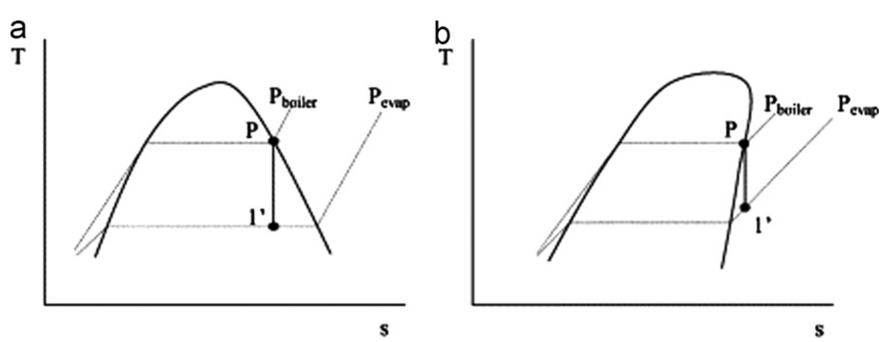

Fig. 3. Expansion process of refrigerant through the primary nozzle [11]. (a) Wet Fluid and (b) Dry Fluid.

should be taken into consideration when choosing a working fluid:

(1) Thermo-physical properties:
- The fluid should have a large latent heat of vaporization in order to minimize circulation rate per unit of cooling capacity.
- The fluid pressure at the generator temperature should not be too high in order to avoid heavy construction of the pressure vessel and to minimize the power required by pump.
- Transport properties that influence heat transfer, e.g., viscosity and thermal conductivity should be favourable.
- Working fluid with smaller value of molecular mass requires comparatively larger ejectors for the same system capacity. The difficulties of constructing small-scale ejector components should be considered. However, higher molecular mass fluid leads to an increase in entrainment ratio and ejector efficiency.

(2) Environmental impact: the fluid should be environmental friendly with relatively low ozone depletion potential (ODP) and global warming potential (GWP).
(3) Safety: the fluid should be chemically stable, non-toxic, non-explosive, non-corrosive.
(4) Economics and availability: the fluid should be low cost and available on the market.

According to Chen et al. [11], working fluid for an eject refrigerator can be categorized as wet vapour and dry vapour as shown in Fig. 3. For wet vapour fluid, its saturated vapour line forms a negative slope in the T–S diagram. For dry vapour fluid, there is no phase change during the expansion process through

the primary nozzle. On the other hand, for wet vapour fluid, small droplets may be formed at the nozzle exit, which may block the effective area and bump into the wall and affect the proper working of ejector. This may be eliminated by superheating the fluid before entering the nozzle. However, the use of superheated motive steam causes a slight decrease in ejector efficiency. Hence, dry vapour is more desirable than wet vapour fluid.

Based on their chemical composition, the working fluids can be classified into the following categories:

(1) Halocarbon group, e.g. R11, R113, R114, R134a, R245ca, R245fa and R152a.
(2) Hydrocarbon group, e.g. methane (R50), ethane (R170), propane (R290), cyclopropane (RC270), butane (R600), isobutene (R600a), and ethylene glycol.
(3) The compound refrigerants, e.g. R407A, R407B, and R410A.
(4) Other refrigerants, e.g. water (R718), carbon dioxide (CO_2) and ammonia (R717).

Table 2 lists some fluids commonly considered for ejector refrigeration systems.

The ozone crisis has caused a major stir in the refrigeration industry and has triggered a critical look at the refrigerants in use. Commercial widely used working fluids, such as CFCs, HCFCs and HFCs, have many advantages compared with natural refrigerants. However, they allow more ultraviolet radiation into the earth's atmosphere by destroying the protective ozone layer and thus contributing to the greenhouse effect and global warming. After the Montreal Protocol in 1987, the intention of governments worldwide was to prohibit the use of ozone depleting refrigerants, which has thus forced refrigeration engineers to search for new fluids.

Hence, many working fluids suggested in previous work for ERS are now, forbidden due to their environmental effect, such as R11, R12, R113, or R114. New refrigerants are now studied, for example, halocarbon compounds (R134a, R152a, R245fa, etc.) and hydrocarbon compounds (R290, R600, R600a), carbon dioxide (R744) and ammonia.

Cizungu et al. [14] simulated the jet refrigerator with various refrigerants including, R123, R134a, R152a and R717. It was found that, with the same ejector, R134a and R512a were suitable for 70–85 °C heat sources and ammonia was suitable for the heat source whose temperature was greater than 90 °C.

An analysis of ejector refrigerator with environmental friendly refrigerants was implemented by Selvaraju et al. [19]. Comparisons of performance of ejector with five working fluids, R134a, R152a, R290, R600a and R717 were made. Among those selected refrigerants, R134a (COP=0.31) proved to yield a better performance, followed by R152a (COP=0.27), R290 (COP=0.25), R600a (COP=0.23) and R717 (COP=0.05) at boiler temperature of 85 °C, condenser temperature of 25 °C and evaporator temperature of 5 °C. A similar experimental analysis on a solar assisted ERS was presented by Nehdi et al. [33] with R134a, R141b, R142b, R152a, R245fa, R290, R600 and R717 as refrigerants. Comparative calculations showed that R717 offered the highest COP (0.408), followed by R152a (COP=0.385), R134a (COP=0.379) and R290 (COP=0.372) at boiler temperature of 90 °C, condenser temperature of 35 °C and evaporator temperature of 15 °C. However, simulation carried out by Roman [37] with low ecological impact refrigerants indicated that R290 (COP=0.66) demonstrated better performance, which followed by R152a (COP=0.58) and R134a (COP=0.56), R600a (COP=0.48) and R600 (COP=0.47) at boiler temperature of 90 °C, condenser temperature of 30 °C and evaporator temperature of 10 °C.

R245fa and R123 were taken as refrigerants by Eames et al. [48] and Yapici et al. [13] in experimental studies on ejector refrigerators, respectively. The results indicated that both R245fa and R123 were practical working fluids for jet-pump refrigeration systems.

ERS using low temperature halocarbon compounds as refrigerants have the advantage that the system can be driven by low-grade heat source (such as solar energy, waste heat and exhaust gas from automobile). A solar-driven ERS with R600a as refrigerant was studied by Pridasawas et al. [39]. And COP of the cooling sub-system was about 0.48. Boumaraf et al. [40]carried out a simulation programme on an ERS, with R142b and R600a as refrigerants. It was demonstrated that R142b produced better performance of the system. As R142b is a heavier fluid than R600a, this further supported the conclusions made by Holton et al. [49] that an ejector performed better with high molecular weight fluid.

Carbon dioxide was considered as a refrigerant prior to and after the turn of the twentieth century. Since the working characteristic and economic availability are not as appealing as fluorocarbons, carbon dioxide had not been widely applied since. Not like those synthetic refrigerants, carbon dioxide is a non-flammable natural substance with zero ozone depletion and a negligible GWP. Therefore, recent research has shown an increasing interest in carbon dioxide as a refrigerant. Owing to the fact that the critical temperature of carbon dioxide is 30.85 °C, while the environmental temperature of a typical summer day may reach 35 °C or more, a supercritical heat rejection temperature is required in the ERS. This requires the system to work under a transcritical CO_2 cycle, which is different from other refrigerants.

Using water as the working fluid for a jet refrigerator provides many advantages. Its extremely high heat of vaporization causes a low circulation rate for given cooling capacity. Therefore, low mechanical power is required for the pump. Water is inexpensive

Table 2
Working fluids for ejector refrigeration system.

Type of refrigerant	Boiling point at 1 atm (°C)	Molecular mass (kg/kmol)	Latent heat at 0 °C (kJ/Kg)	Global warming potential (GWP)	Ozone depletion potential (ODP)	Wet/dry vapour	Reference
R123	27.9	152.39	176.8	0.02	0.016	Dry	[13–18]
R134a	−26.1	102.03	190.9	0.26	0.020	Wet	[8,10,14,19–31]
R141b	32.1	116.9	129.4	1.2	0	Dry	[12,20,24,32–36]
R142b	−9.2	100.5	215	0.36	0.06	Dry	[20,29,39,64,107]
R152a	−24.0	66.05	324.2	2.8	0	Wet	[19,27,33,37,38]
R245fa	59.5	134	196.7	950	0	Dry	[13,32,47,56,57,60,113]
R290	−42.1	44.1	357.2	3	0	Wet	[19]
R600	−0.5	58.12	385.6	20	0	Dry	[33,37]
R600a	−0.5	58.12	374.3	< 10	0.043	Dry	[19,37,39,40]
R718b (water)	100.0	18.02	2257.0	0	0	Wet	[5–7,41–44]
R717(ammonia)	−33.34	17.03	1369	0	0	Dry	[9,14,45–47]
R747 (CO_2)	−78.5	44.0	571.1	1	0	Wet	[89–93,108]

and has minimal environment impact (zero ozone depletion and global warming potential). However, there are some drawbacks. Using water as a refrigerant limits the cooling temperature to above 0 °C and the system must be under vacuum condition. Moreover, water has very large specific volume at typical evaporator conditions and to minimize the pressure loss, pipe diameter must be large to accommodate the large volume flow rate. Experiments show that a steam-jet refrigerator requires a boiler temperature between 80 °C and 140 °C. Recently, Butterworth et al. [41] studied the feasibility of using high-pressure water as driving fluid in an ERS. With the high-pressure water available from vertical pipelines in deep mine shafts, he concluded that the system performance improved as the motive water pressure increased.

The industrial and heavy-commercial sectors were very satisfied with ammonia, even though ammonia is toxic. The advantages of ammonia over the other refrigerants are its low cost, high COP (and thus lower energy cost), more favourable thermodynamic and transport properties, greater detectability in the event of leakage, and no effect on the ozone layer. The major drawback of ammonia is its toxicity, which makes it unsuitable for domestic use. Thus, ammonia is predominantly used in food refrigeration facilities, low-temperature refrigeration in the pharmaceutical and other process industries.

3. Recent development of ejector models

Ejector refrigeration systems were first invented by Sir Charles Parsons around 1901 for removing air from a steam engine's condenser. It was later used in the first steam jet refrigeration system by Maurice Leblanc et al. [50] in 1910. Since then, considerable efforts have been concentrated on the enhancement and refinement of ERS.

3.1. Single phase models

A one dimensional model described by Keenan et al. [51] in 1950 was the first application of continuity, momentum and energy equation in ejector design principle. This model has been used as a theoretical basis in ejector design since then. Keenan's model, however, cannot predict the constant-capacity characteristic and was later modified by Munday and Bagster [52]. Based on their theory, it is assumed that the primary fluid flows out without mixing with the secondary fluid immediately induces a converging duct for the secondary fluid. This duct acts as a converging nozzle such that the secondary flow is accelerated to a sonic velocity at some place, known as effective or hypothetical throat. After that both fluids mix with a uniform pressure.

Eames et al. [44] studied a small-scale steam-jet refrigerator and presented a theoretical model that included irreversibilities associated with the primary nozzle, the mixing chamber and the diffuser. This model was based on constant-pressure mixing process, but without considering the choking of the secondary flow. In order to take this in to account, Huang et al. [36] presented a one-dimensional critical model (double-chocking) by assuming that mixing of two streams occurs inside constant area section with uniform pressure. The model was experimentally verified with 11 different ejectors using R141b as the working fluid. In order to simplify the model, more models [53] were proposed to calculate the performance of ERS. In these models, the thermo-physical and transportation properties need to be obtained from data base, which limits their application. Zhu et al. [54] proposed an ejector for a real time control and optimization of an ejector system, which was based on one-dimension analysis. Though the model is simplified, the expressions were more complex and some parameters needed to be determined experimentally. In order to give a more accurate prediction of the ejector performance in the mixing chamber, Yapici and Ersoy [18] derived a local model based on constant-area mixing process. The ejector consisted of a primary nozzle, a mixing chamber in cylindrical structure and a diffuser. Compared with a similar model designed by Sun and Eames [55] under same operating temperatures, Yapici's model showed better COP. Elakhdar et al. [20] developed a mathematical model in order to specifically design a R134a ejector and predict the performance characteristics over different operating conditions. Simulation results showed that the present model data were in good agreement with experimental data in the literature with an average error of 6%. A constant-area 1-D model was recently presented by Khalil et al. [30]. Governing equations were developed for the ejector's three different operating regimes, supersonic regime, the transition regime and the mixed regime. Environmental friendly refrigerants were used as working fluids in the simulation. Results were compared with that of experimental data available in the literature, and good agreement was demonstrated.

All the above models are based on ideal gas assumption which does not reflect the actual process occurring in the ejector. Rogdakis and Alexis [56] improved the model proposed by Munday and Bagster [52] by using the thermodynamic and transportation properties of real gases. When considering the friction losses, a constant coefficient was assumed to simplify the model. However, the friction losses were closely related to the velocity, and the velocity varied considerably along the ejector. Taking this into account, Selvaraju and Mani [19] developed a model based on Munday and Bagster's theory for critical performance analysis of the ejector system. This model applied an expression to describe the friction losses in the constant area section. A 1-D model avoiding the ideal gas assumption was proposed by Grazzini et al. [57]. Heat exchanger irreversibilities were taken into consideration, and real gas behaviour was simulated. A comparison between different refrigerants was presented and R245fa was selected as a working fluid. However, validation with experimental data in literature was not available. In order to check the validity of the ideal gas assumption, Grazzini et al. [58] evolved another model with the key concept of metastable state. To set the border for metastable region, a spinodal curve was introduced. The modelling results were compared with experimental data. The author concluded that in order to avoid complexity, the metastable behaviour of steam can be implemented in a single 1-D model giving stable results.

3.2. Two phase models

The abovementioned models are based on the assumption that the flow in the ejector is in compressible single phase and recompression occurs across a normal shock wave. However, under many real applications, phase change can occur and a condensation shock may develop. Thus, some researchers are engaged in ejector simulation with two-phase flow. By introducing dryness of the fluid in the calculation of the specific volume, enthalpy and entropy, Sherif et al. [59] derived an isentropic homogeneous expansion/compression model to account for phase change due to expansion, compression and mixing. In this model, the primary fluid was a two-phase mixture and the secondary fluid was either a sub-cooled or saturated liquid having the same chemical composition as the primary fluid. Cizungu et al. [45] derived a two-phase thermodynamic model to calculate the entrainment ratio. This model can be used both for single-phase and two-phase ejector with single component or two components working fluids. He et al. [60] investigated the usefulness of a multivariate grey prediction model, which incorporated grey relational analysis to

predict the performance of ERS. The importance of influencing variables was first evaluated, and then the variables were ranked according to the grey relational method. It also compared the performance of the combined grey model with that of conventional one-dimension theory model as well as experimental data. The simulation results showed that the grey system theory can be used to analyze the ERS.

3.3. CFD models

Despite the remarkable progress, made in thermodynamic modelling, these models were unable to reproduce the flow physics locally along the ejector. It is the understanding of local interactions between shock waves and boundary layers, their influence on mixing and re-compression rate that will produce a more reliable and accurate design, in terms of geometry, refrigerant type and operation conditions. Computational Fluid Dynamics (CFD) modelling can provide more accurate simulations of the ejector in accordance with experiment results. Early CFD studies can be traced back to late 1990s. However, they failed to overcome some of the fundamental problems, especially regarding the simulation of shock-mixing layer interaction and ejector operation under different working conditions. Compressibility or turbulence was hardly taken into consideration. Even when turbulence was considered, only *k*-epsilon based models were used. No experimental validations or justification, except for CPU cost were carried out.

Recently, Rusly et al. [32] stimulated the flow through an R141b ejector by using the real gas model in the commercial code, FLUENT. The effects of ejector geometries on system performance were investigated numerically. The CFD results were validated with experimental data and good agreement was found. The selection of correct turbulence model plays an important role in predicting the mixing process in the ejector for CFD studies. Turbulence effects in the ejector have been modelled using the standard *k*-epsilon turbulence model by Scott [61] using CFD. The CFD results were later verified with an experimental investigation of an ejector with R245fa as a working fluid [62]. Comparisons were made between results from experiments, CFD model and a theoretical 1-D model by Ouzzane and Aidoun [63]. It was concluded that CFD model provided better agreement (difference of less than 16%) than 1-D model.

Aiming at validation the choice of a turbulence model for the computation of supersonic ejectors in refrigeration applications, Bartosiewicz et al. [64] compared experimental distribution data with results of simulation using different turbulence models. However, the choice of air as working fluid and other test conditions were not very in accordance with cooling cycles. Later they extended their work using R142b as refrigerant et al. [65]. With consideration of shock-boundary layer interactions, this ejector model contributed to the understanding of the local structure of the flow and demonstrated the crucial role of the secondary nozzle for the mixing rate performance. Pianthong et al. [5] employed the CFD with realizable *k*-epsilon turbulence model to predict the flow phenomena and performance in steam ejectors with application in refrigeration system. The result indicated that CFD can predict ejector performance very well and reveal the effect of operating conditions on the effective area that was directly related to its performance. In order to consider the sensitivity of the turbulence model over several conditions, Hemidi et al. [66] carried out CFD analysis of a supersonic air ejector with single and two phase operation. Entrainment ratio based on *K*-epsilon model and k-ω-sst model were compared with experimental data. The results demonstrated that even with the same prediction level, both models could provide very different local flow structures.

3.4. Non-steady flow models

Since the mid-1990s, some researchers have focused on the theory and implementation of non-steady or pressure-exchange ejector. Compared with conventional steady ejectors, non-steady ejectors allow energy transfer between two directly interacting fluids but maintaining them separable. By utilizing the reversible work of pressure forces acting at fluid interfaces between primary flow and secondary flow, non-steady ejectors have the potential of much greater momentum transfer efficiency. Recently, Hong et al. [67] presented a novel thermal driven rotor-vane/pressure-exchange ERS. Unlike other pressure-exchange ejectors which had canted primary nozzles on their rotors, the rotor of this ejector had vanes directly on it and a primary nozzle separated from it. Computational study and experimental work were included in order to optimize the ejector geometry. However, the concentrations were only placed on the overall shape of rotor vane, without considering Mach number of incoming flow, the geometry in the interaction zone and the diffuser geometry.

Ababneh et al. [68] studied the effects of the secondary fluid temperature on the energy transfer in a non-steady ejector with a radial-flow diffuser. The flow field was analyzed at Mach numbers 2.5 and 3.0, with a range of temperatures from -10 °C to 55 °C. The results revealed that the actual energy transfer to the secondary fluid, which included the effects of irreversibilities, decreased with the increase in ambient temperature. However, due to mechanical difficulties, the experimental work was halted, only numerical simulation was presented. Gould et al. [29] carried out theoretical analysis of a steam pressure exchange (PE) ejector in automotive air conditioning (AC) system. Waste heat from the engine of vehicle was utilized as the main heat source. Comparisons were made between a conventional R134a AC system and the steam PE ejector AC system at idling and 50 mph conditions. The results showed that the steam PE ejector system consumed at least 68% less energy than R134a AC system. And COP of PE ejector AC system was 2.5–5.5 times that of R134a AC system at both conditions. However, the theoretical data were not verified with experimental results.

Table 3 lists the references with different model types and their key simulation results.

4. Recent development in ejector geometric optimization

In order to make the ejector system more economically attractive, a number of researches have investigated the optimization of the ejector geometry on system performance.

4.1. Area ratio

An important non-dimensional factor affecting ejector performance is the area ratio γ_A between primary nozzle and constant area section. It is known that flow emerges from the primary nozzle and maintains its definition as primary fluid for some distance. The secondary fluid is entrained into the region between the primary fluid and the ejector wall. If an ejector of fixed primary pressure, secondary pressure and nozzle geometry is considered, increasing the mixing section area will result in a greater flow area for the secondary stream. The entrainment ratio will therefore increase but since the compression work available from the primary flow is unchanged, the ejector is unable to compress to higher discharge pressures. In this case, according to Varga et al. [42] increasing γ_A increases entrainment ratio and decreases the critical back (condenser) pressure and therefore an optimal value should exist, depending on operating conditions.

Table 3
Working conditions and simulation results for selected models.

Reference	Simulation model	Refrigerant	Evaporator temperature (°C)	Condenser temperature (°C)	Boiler temperature (°C)	System performance	
						COP	Condition
Eames et al. [44]	Constant pressure model with irreversibilities	water	5:10	26:37	120:140	0.239	$T_b=120$ °C, $T_c=27$ °C, $T_e=5$ °C, $\gamma_A=102$
Yapıcı and Ersoy [18]	Constant area model	R123	5	30	60:100	0.295	$T_b=100$ °C, $T_c=30$ °C, $T_e=5$ °C, $\gamma_A=11.46$
Khalil et al. [30]	Constant area model takes into account of R134a vapour state and friction loss	R134a	6:10	25:40	65:85	0.355	$T_b=70$ °C, $T_c=35$ °C, $T_e=10$ °C, $\gamma_A=0.838$
Grazzini et al. [57,58]	1-D model with real gas behaviour and heat exchanger irreversibilities Diffuser design on a modified CRMC criterion accounts for friction loss	R245fa	12	35	115	0.325	$T_b=115$ °C, $T_c=35$ °C, $T_e=12$ °C
Cizungu et al. [45]	A two-phase thermodynamic model takes into account the duct effectiveness, wall friction, momentum loss, ejector geometry, shock waves (with no assumption of constant area/pressure mixing)	R11	3.5:8.5	30:35	80:130	0.415	$T_b=90$ °C, $T_c=30$ °C, $T_e=8.5$ °C, $\gamma_A=6$
Pianthong et al. [5]	CFD modelling	water	5:15	15:35	120:140	0.42	$T_b=120$ °C, $T_c=30$ °C, $T_e=10$ °C
Aidoun and Ouzzane [12]	1-D model accounts for changes in refrigerant properties with the flows axial position	R141b	5	NA	75	0.322	$T_b=75$ °C, $T_e=5$ °C,
Pc=120:380 Kpa							
Boumaraf and Lallemand [40]	Constant area model includes a correlation of the ejector entrainment ratio established in different operating conditions	R142b	10	35	120:130	0.128	$T_b=120$ °C, $T_e=10$ °C
		R600a	10	35	120:130	0.089	$T_b=120$ °C, $T_e=10$ °C

Yapici et al. [15] studied the performance of R123, using six configurations of ejector over a range of the ejector area ratio from 6.5 to 11.5. It was concluded that the optimum area ratio increased approximately linearly with generator temperature in the ranges of 83–103 °C. Instead of using water-cooled condenser, Jia et al. [21] presented an experimental investigation on air-cooled ERS using R134a with 2 kW cooling capacity. Replaceable nozzles with varying ejector area ratios from 2.74 to 5.37 were used, and the best system performance was shown for area ratio from 3.69 to 4.76. Cizungu et al. [45] modelled a two-phase ejector with ammonia as working fluid, and found out a quasi linear dependence between γ_A and the driving pressure ratio (pressure ratio of boiler to condenser). This result was suitable for the rough draft of sizing and operational behaviour of the refrigerator.

Area ratio, however, can be identified as a single optimum that would bring the ejector to operate at critical mode for a given condenser temperature. Obviously, this would require different ejectors for different operating conditions. In order to overcome this problem, a new feature – a spindle – was implemented and tested numerically and experimentally by Ma et al. [6] and Varga et al. [69]. By changing the spindle position, the area ratio γ_A can be changed. As the spindle tip travels forward, the primary nozzle throat area decreases, and consequently γ_A increases. CFD simulation was carried out by Varga et al. [70] to analyze the effect area ratio on the ejector performance. The authors indicated that ejectors with area ratios varying from 13.5 to 26.4 could achieve entrainment ratios from 0.18 to 0.38. They also pointed out that by changing the spindle position, an optimal γ_A can be achieved with a single ejector. Experimental investigation of this spindle system was carried out by Ma et al. [6] using water as refrigerant. The results showed that when spindle position was 8 mm inwards the mixing chamber, an optimum entrainment ratio of 0.38 could be achieved, which were less than the maximum value of Varga's CFD modelling [70] at almost same designed working conditions. The group [69] later summarized and compared the experimental results with CFD data. It was concluded that CFD and experimental primary flow rates agreed well, with an average relative error of 7.7%.

Table 4 shows the various studies on the ejector's area ratio with different working conditions

4.2. Nozzle exit position (NXP)

The nozzle exit position (NXP) inwards or outwards the mixing chamber is known to affect both the entrainment and pressure lift ratio performance of ejectors. In the experimental studies [7,10,16] and CFD simulations [32,64,71–74], it was demonstrated that moving the nozzle exit into the mixing chamber reduced COP and cooling capacity. Recently, Eames et al. [48] found a clear optimum of the entrainment ratio (40% increase) at 5 mm from the entrance of the entrainment chamber. In this case the ejector tail was designed by the constant momentum rate change (CMRC) method and R245fa was used as working fluid. Similar conclusions can be found from numerical investigations carried out by Varga et al. [70] and Zhu et al. [75]. CFD modelling results from Varga et al. [70] indicated that an optimum entrainment ratio of 0.33 can be achieved when NXP was 60 mm downstream. Zhu et al. [75] reported that the optimum NXP was not only proportional to the mixing section throat diameter, but also increased as the primary flow pressure rises. The authors also pointed out that the ejector performance was very sensitive to the converging angle θ of the mixing section. When NXP was within its optimum range, the optimum θ was in the range of 1.45–4.2°. A relative larger θ was required to maximize ejector performance when the primary flow pressure rised.

Table 4
Results on the ejector's area ratio with different working conditions.

Reference	Method	Working fluid	Area ratio	COP	T_e (°C)/P_e	T_c (°C)/P_c	T_b (°C)/P_b	Conclusion
Yapici et al. [15]	Experiment	R123	6.5:11.5	0.29:0.41	10 °C	125 kpa	83:103 °C	(1) The optimum area ratio nearly linearly increased with the generator temperature in the studied range. (2) For a given ejector area ratio, there existed an optimum generator temperature at which maximum COP was obtained from the ejector refrigeration system.
Jia et al. [21]	Experiment	R134a	2.74:5.37	0.14:0.35	4.43 bar	7.8 bar	17.5:16.5 bar	(1) Optimum area ratio was found between 3.69 and 4.76, with maximum COP in the range of 0.24–0.30. (2) For given operating conditions, the cooling capacities were related to area ratios as well as nozzle diameters while COPs depended only on area ratios.
Cizungu et al. [45]	Numerical modelling	Ammonia	4:12	0.24:0.34	4.5 °C	32 °C	80:130 °C	There exited a quasi linear dependence between γ_A and the driving pressure ratio (pressure ratio of boiler to condenser).
Varga et al. [69,70]	CFD modelling	Water	13.5:26.4	0.18:0.38	10 °C	4.25: 5.63 kpa	70.1:101 kpa	By changing the spindle position, the effective nozzle area can be adjusted and an optimal γ_A can be adjusted with a single ejector
Ma et al. [6]	Experiment	Water		0.17:0.32	10 °C	25 mbar: 65 mbar	84:96 °C	

In contrast, CFD analyses of Rusly et al. [32] and Pianthong et al. [5] showed that NXP only had a small influence on entrainment ratio. In the first case, a 20% variation in NXP was considered in an ejector using R141b as working fluid. Compared to the base model, moving the nozzle towards the constant area section caused λ to decrease, while moving it in the other direction λ remained practically unchanged. The authors claimed that the optimum NXP of 1.5 diameters of the constant area section produced better performance. Pianthong et al. [5] varied NXP in the range from -15 to 10 mm from the mixing chamber inlet. The entrainment ratio increased slightly as NXP was moved further from the inlet section.

Optimum primary nozzle position or converging angle cannot be predefined to meet all operating conditions. When the operating conditions are different from the design point, the NXP should be adjusted accordingly to maximize the ejector performance. An ejector with movable primary nozzle can provide a flexible NXP when the conditions are out of the design point. This was first presented by Aphornratana and Eames [76]. Recently Yapici et al. [13] carried out an experimental investigation on an ejector refrigerator with movable primary nozzle. The author concluded that the optimum primary nozzle exit should be -5 mm from the mixing chamber inlet.

Due to the varying nature of the operation conditions as well as the different ejector geometries, no general agreement can be achieved among various researches.

4.3. Primary nozzle diameter

The relationship between primary nozzle diameter and the boiler temperature was reported by Cizungu et al. [45]. Using ammonia as working fluid, the author stated that the optimum primary nozzle diameter decreased with increase in the boiler temperature. Similar results were obtained by Sun [16] with an ejector driven by the working fluid R123. Chaiwongsa et al. [10] analyzed the effects of various nozzle outlet diameters D_{nt} (D_{nt}=2 mm, 2.5 mm and 3 mm) of a motive nozzle on the system performance. The Nozzle with outlet diameter of 2 mm was found to yield the highest COP.

4.4. Constant area section length and diffuser geometry

Constant area section length is commonly believed to have no influence on the entrainment ratio [26,40]. However, Pianthong et al. [5] reported that the critical back pressure increased with Lm and thus allowed to operate the ejector in double chocking mode in a wider range of operating conditions.

As seen from Fig. 1, a thermodynamic shock wave can cause a sudden fall in Mach number as the flow changes from supersonic ($Ma > 1$) to subsonic ($Ma < 1$). This process results in a fall in total pressure and this effect reduces the maximum pressure lift ratio, which a conventional ejector refrigerator can achieve. With the aim to overcome this shortfall, Eames et al. [77] developed the constant rate of momentum change (CRMC) method to produce a diffuser geometry that removes the thermodynamics shock process within the diffuser at the design-point operating conditions. Theoretical results described in this paper indicated significant improvements in both entrainment ratio and pressure lift ratio, above those achievable from ejector designed using conventional methods. Experimental data were presented by Worall et al. [78] that supported the theoretical findings.

5. Recent development in ejector performance improvement

The ability of making use of renewable energy and the advantages of simplicity in construction, installation and maintenance make ERS more cost-effectively competitive compared with other refrigeration system. The system performance for ERS, however, is relatively low. Hence, the engineers and researchers are making efforts to improve system efficiency for ERS. Past decade has seen many research innovations of enhancing system performance, including reduction of the mechanical pump work in ERS, utilization of the special refrigerants, and utilization and storage of available renewable energy. Many research groups have widely carried out theoretical calculations, computer simulations and experimental works in these areas.

5.1. Ejector refrigeration system without pump

The pump, with the function to convey liquid condensate in the condenser back to the generator, is the only moving part in the ERS. This equipment, however, not only requires additional mechanical energy, but also needs more maintenance than other parts. Hence, many researchers have tried to utilize other methods to eliminate those shortcomings.

5.1.1. Gravitational ejector

Kasperski et al. [79] presented a gravitational ERS (as shown in Fig. 4) in a simulation model. Unlike the pump version ejector refrigerator, the heat exchangers are placed at different levels. Thus, with the help of the refrigerant hydrostatic pressure, the vertical arrangement of the heat exchangers enables pressure differences between the exchangers to be equalized. The lowest pressure in the refrigerator installation was obtained in the evaporator. It caused the inflow of liquid to the highest installation level. The highest pressure was obtained in a steam generator, which forces the lowest liquid level.

The limitation of this system lies in its requirement of great height differences and the length of pipe work, which increases friction and heat losses. Therefore, the conception of the gravitational refrigerator (Fig. 5a) was later developed into a rotating refrigerator (Fig. 5b) by Kasperski et al.[80]. With lager accelerations of rotary motion, this roto-gravitational refrigerator significantly decreased the size of the gravitational refrigerator. However, the author only proposed a mathematical model, no experimental results were presented.

5.1.2. Bi-ejector refrigeration system

A schematic view of a solar-powered bi-ERS designed by Shen et al. [81] is shown in Fig. 6. In this system, an ejector (injector) replaces the mechanical pump to promote pressure of the liquid condensate and conveys the condensate back to the generator. Ideally, the system will lead to zero electricity consumption.

The authors studied the performance of this system with different refrigerants using numerical modelling. The result showed that the overall COP of the system was mainly affected by the gas–gas ejector entrainment ratio in the refrigeration loop. Compared with other refrigerants, under the same operating conditions, the gas–liquid ejector (injector) entrainment ratio of R718 was relatively high. However, the best overall system COP achieved was 0.26 using R717 as the refrigerant.

In the proceeding work, Wang and Shen [82] took considerations of the effect of injector structures on the system performance. The real fluid's thermal properties were considered in the new injector thermodynamic model. The authors concluded that with increasing generator temperature, the entrainment ratio of the injector and the thermal efficiency of the solar collector were reduced, whilst the entrainment ratio of the ejector and COP of bi-ER sub-system were improved. The overall COP of the system reached an optimum value of 0.132.

5.1.3. Ejector refrigeration system with thermal pumping effect

An ERS that utilizes a multi-function generator (MFG) to eliminate the mechanical pump was presented by Huang and Wang [83,84]. The MFG serves as both a pump and a vapour generator.

The schematic diagram of the ejector cooling system with multi-function generator (ECS/MFG) is shown in Fig. 7. There were two generators in the ECS/MFG. Each generator consisted of

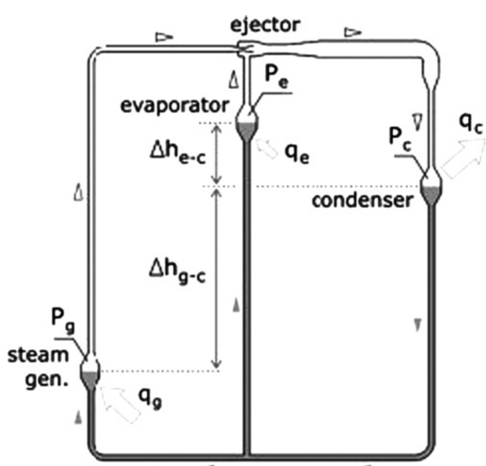

Fig. 4. Schematic diagram of a gravitational ejector refrigerator [79].

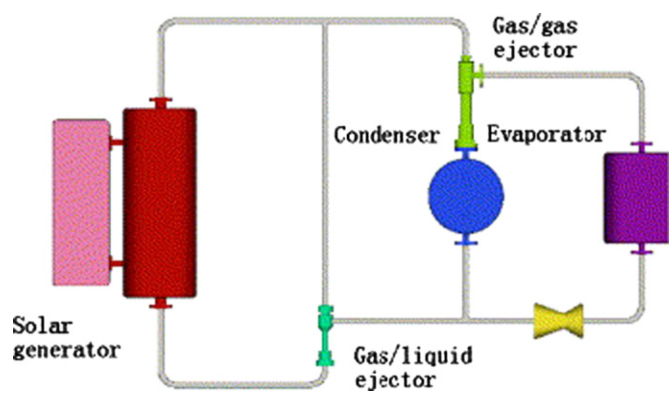

Fig. 6. Schematic diagram of solar-powered bi-ERS [81].

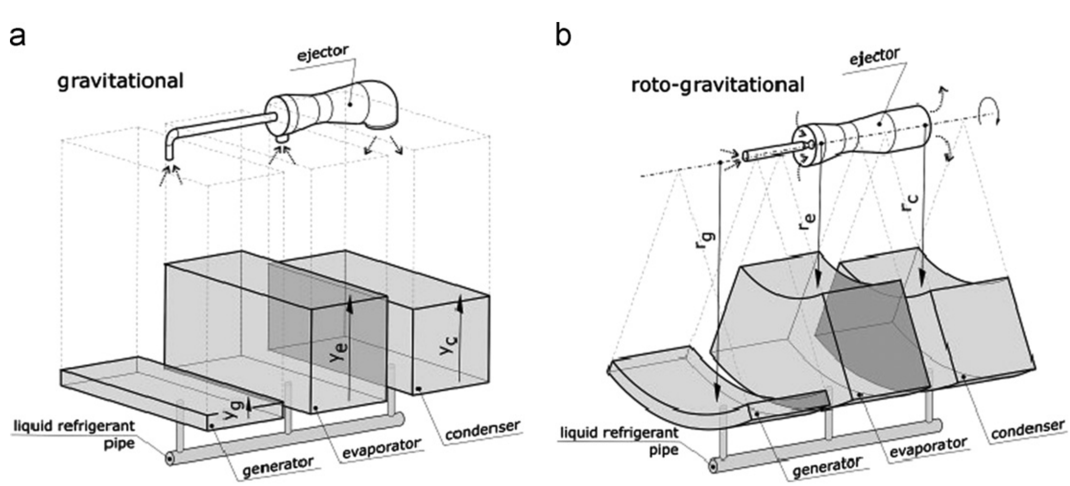

Fig. 5. Schematic diagram of liquid refrigerant levels in gravitational (a) and roto-gravitational (b) ejector refrigerators [80].

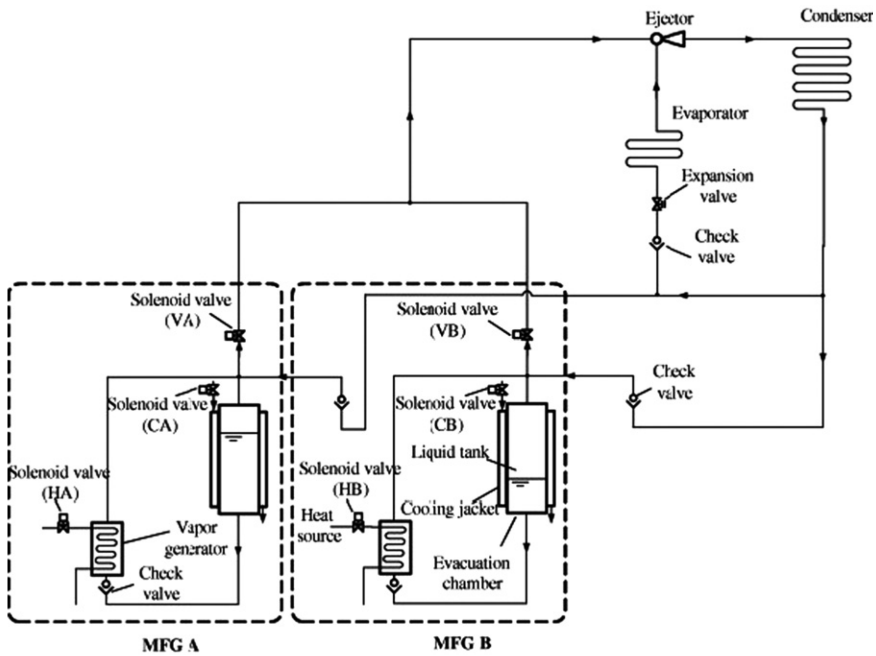

Fig. 7. Schematic diagram of ejector cooling system with multi-function generator [83].

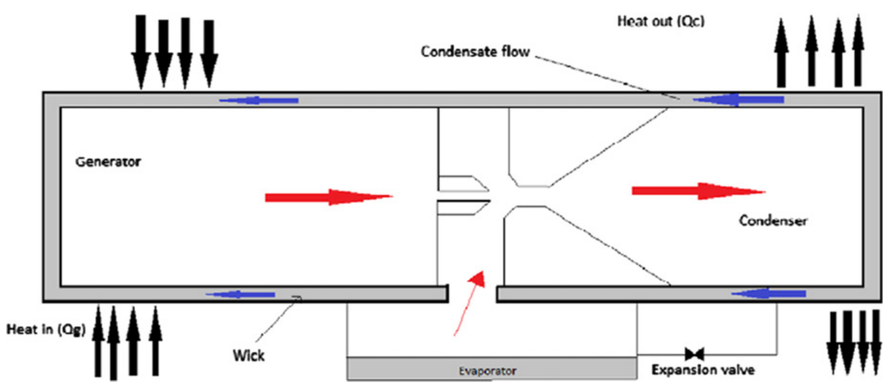

Fig. 8. Schematic diagram of heat pipe/ejector refrigeration system [85].

a vapour generator and an evacuation chamber. The vapour generator was a heat exchanger like a conventional boiler for pressurizing and to generating vapour. The evacuation chamber was composed of a cooling jacket and a liquid holding tank. The cooling jacket provided a cooling effect to depressurize the generator in order to intake the liquid from condenser. Detailed system description can be referred to Huang and Wang [83,84].

This system makes use of the pressure change in the generator to create backflow of liquid condensate. However, the system is composed of too many elements, which will lead to unavoidable consumption of available thermal energy.

5.1.4. Heat pipe and ejector cooling system

Integration of the heat pipe with an ejector will result in a compact and high performance system, which does not require additional pump work. This system can also utilize solar energy or hybrid sources and so reduces the demand for electricity and thus fossil fuel consumption.

The basic cycle of the heat pipe/ERS is shown in Fig. 8. The system consists of a heat pipe, ejector, evaporator and expansion valve. The low potential heat is added to the system in the generator section. Then the working fluid evaporates and flows

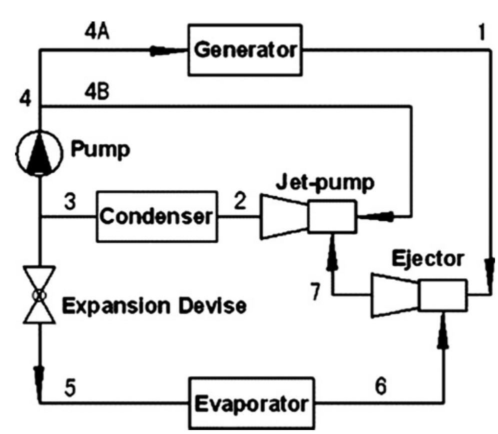

Fig. 9. Schematic diagram of an ejector refrigeration system with additional jet pump [25].

through the primary nozzle of the ejector. Therefore it expands and contributes to the decrease of the pressure in the evaporator. Thus, the refrigeration cycle can be completed. In the condenser, some of the working fluid was returned to the generator by the

wick action, while the remainder was expanded through the expansion valve to the evaporator. Unlike other vapour compression refrigeration system, which is powered by mains electricity generated by large plants, the heat pipe/ERS does not require any electricity input.

With the aim of finding the optimum operating conditions for a heat pipe/ERS, Ziapour et al. [85] carried out an energy and exergy analysis based on the first and second laws of thermodynamics. The simulation results were compared with available experimental data from literature for steam ejector refrigerator. The results showed that COP could reach about 0.30 at generator temperature of 100 °C, condenser temperature of 30 °C and evaporator temperature of 10 °C. The authors also indicated that the maximum heat pipe cooling capacity could be obtained for large heat pipe diameters, near the small heat pipe lengths.

5.2. Ejector refrigeration system with multi-components

Although the single stage ERS is simple, it is difficult to keep the system running at optimum conditions due to the variation of working conditions. Ambient temperatures above design conditions or lower generator temperature often lead to operational difficulties. Attempts have been made to solve this problem by using multi-components ejectors.

5.2.1. Ejector refrigeration system with an additional jet pump

Yu et al. [25] proposed a ERS with an additional liquid–vapour jet pump, as shown in Fig. 9. This additional jet pump is applied to entrain the mixing vapour from the ejector, which acts as secondary flow for jet pump. In this case, the backpressure of the ejector can be reduced by the jet-pump, and then the entrainment ratio and COP of the system could be increased.

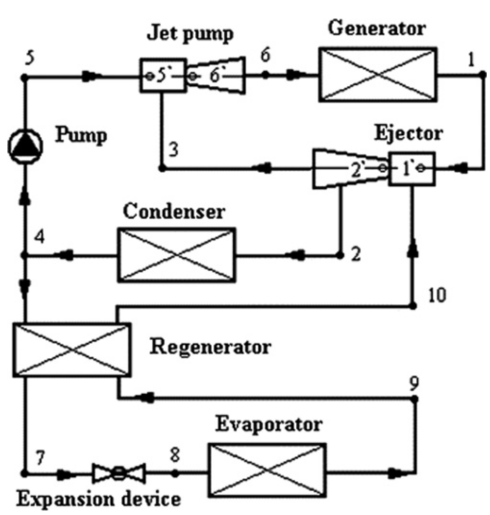

Fig. 10. Schematic diagram of a refrigeration system with additional jet pump [86].

Simulation results showed that, compared with conventional ERS at same working conditions, the COP of NERS was increased by 57.1% and 45.9%, with R152a and R134a, respectively as refrigerants.

The group [86] later presented another system with similar configuration (as shown in Fig. 10). In this system, the auxiliary jet pump was designed to accomplish the effects of both entrainment and regeneration. Different from conventional ERS, the exhaust of the ejector in this system was divided into two parts. One part was discharged at the normal condenser pressure, another part was discharged at a higher pressure than the condenser pressure, and thus this part with higher temperature was rejected as heat for regeneration. Compared with the conventional system, the simulation results showed that the COP of this system increased from 9.3% to 12.1% when generator temperature was in a range of 80–160 °C, the condensing temperature was in a range of 35–45 °C and the evaporating temperature was fixed at 10 °C.

5.2.2. Multi-stage ejector refrigeration system

An example of the multi-stage ejector refrigeration system arrangement is shown in Fig. 11. Several ejectors are placed in parallel before the condenser. One ejector operates at a time and the operation of each ejector is determined by the condenser pressure. Ejector 1 operates when the condenser pressure is below P_{c1}; ejector 2 operates at a condenser pressure between P_{c1} and P_{c2}; and ejector 3 operates at a condenser pressure between P_{c2} and P_{c3}. This arrangement was proposed by Sokolov and Hershgal [87]. However practical work was not available.

5.2.3. Multi-evaporator compression system

Multi-evaporator compression systems (MECS) are generally used in transport refrigeration applications. Kairouani et al. [27] studied a multi-evaporator refrigeration system utilizing ejector for vapour pre-compression. As shown in Fig. 12, the ejectors are

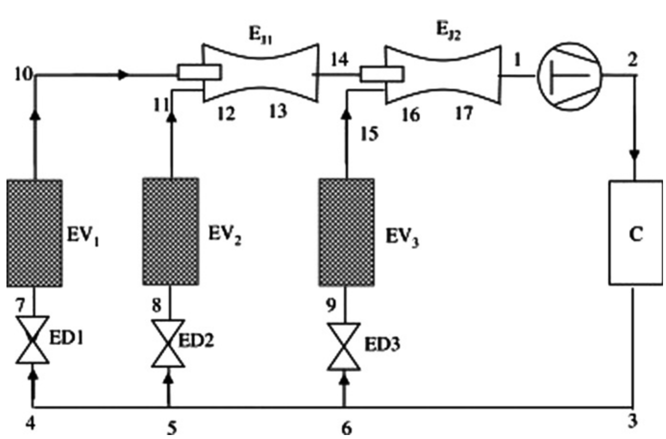

Fig. 12. Schematic diagram of a multi-evaporators compression system [27].

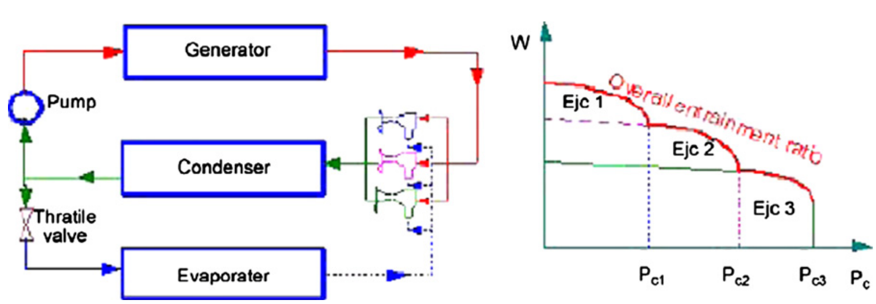

Fig. 11. Schematic diagram of multi-stage ERS [87].

positioned at the outlets of evaporators, which can increase the suction pressure. In the diffuser, the kinetic energy of the mixture is converted into pressure energy. The specific work of the compressor is reduced and then the COP of the system is improved. A comparison of the system performances with environment friendly refrigerants (R290, R600a, R717, R134a, R152a, and R141b) is made. R141b proved to give the most advantageous COP among all working fluids.

Liu et al. [88] presented three different configurations for two-evaporator refrigeration cycle. As shown in Fig. 13 the working principles of series (Fig. 13a) and parallel (Fig. 13b) systems can be easily recognized from the schematic views. The combined circulatory cross-regenerative thermal system (Fig. 13c) is an improvement of the hybrid circulatory system, where the evaporators of the cooling chamber and freezing chamber are in parallel. The pressure at the connector of the three ejectors and the power consumption were measured, and the performances of the three different connection forms for compressor-jet mixing of the refrigeration cycle were compared. Results showed that for the compression–injection crossed-regenerative hybrid refrigeration

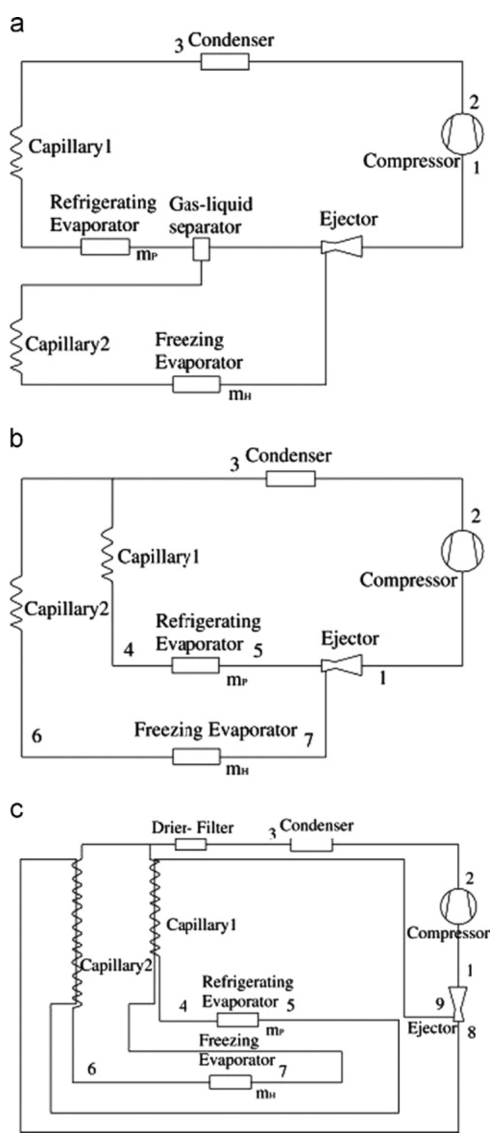

Fig. 13. Schematic diagram of (a) two-evaporator refrigeration cycle in series hybrid system, (b) two-evaporator refrigeration cycle in parallel hybrid system, (c) two-evaporator refrigeration cycle in parallel and crossed-regenerative hybrid systems [88].

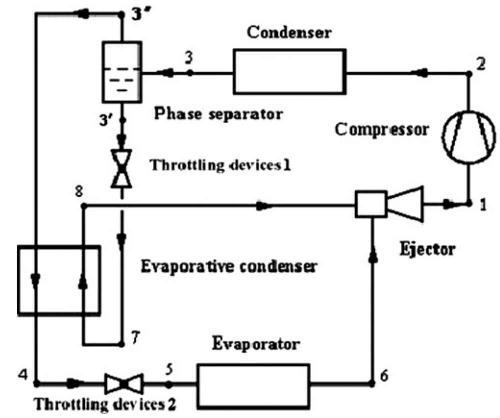

Fig. 14. Schematic diagram of autocascade refrigeration cycle with an ejector [28].

cycle system, loss of heat in the throttle processing was decreased effectively by an ejector. Energy consumptions of the first two prototypes were 0.775 kWh/day and 0.748kWh/day, which were higher than the traditional prototype. The power consumption can be reduced to 0.655 kWh/day for the third one, which was 7.75% lower than the traditional prototype.

Autocascade refrigeration system can use only one compressor to obtain lower refrigerating temperature between $-40\,^{\circ}$C and $-180\,^{\circ}$C. In order to reduce the throttling loss generated by throttling devices, an ejector is introduced to the system to recover the kinetic energy in the expansion process. Yu et al. [28] applied an ejector in autocascade refrigerator with refrigerant mixture of R23/R134a. As shown in Fig. 14, the ejector is set between the evaporative, condenser and the evaporator. Thermodynamic analysis showed that the system employed with an ejector had merits in decreasing the pressure ratio of the compressor as well as increasing COP. With condenser outlet temperature of 40 °C, evaporator inlet temperature of $-40.3\,^{\circ}$C, and the mass fraction of R23 was 0.15, the COP was improved by 19.1% over the conventional autocascade refrigeration cycle.

5.3. Transcritical ejector refrigeration system

Different from other ERS whose refrigerants are working in their subcritical cycle, the refrigerant of transcritical ERS (TERS) operates in transcritical process. Characteristic for the process is heat rejection in the supercritical region, introducing a gliding temperature instead of condensation at constant temperature. Compared with an ERS, the TERS has a higher potential in making use of the low-grade thermal energy with gradient temperature due to a better matching to the temperature glide of the refrigerant.

Yu et al. [89] carried out a theoretical study of a transcritical ERS (TERS) with R134a as working fluid. The schematic diagram is shown in Fig. 15. The study calculation model for the ejector is the constant-pressure mixing model. The generating temperature ranged from 60 to 100 °C with a pressure range of 6–10 MPa. The numerical results indicated that COP of TERS were between 0.35 and 0.75, almost double that of conventional ERS, with generator temperature at 80 °C, evaporator temperature in the range of 10–15 °C and the condensing temperature in the range of 30–40 °C. The authors conclude that the higher working pressure in the TERS resulted in a more compact system. However, no experimental verification is available.

Similarly, a number of studies [90–93] have concentrated on TERS with CO_2 as refrigerant. Li and Groll [90] investigated theoretically the performance of transcritical CO_2 refrigeration cycle with ejector-expansion device (as shown in Fig. 16). This system incorporated a

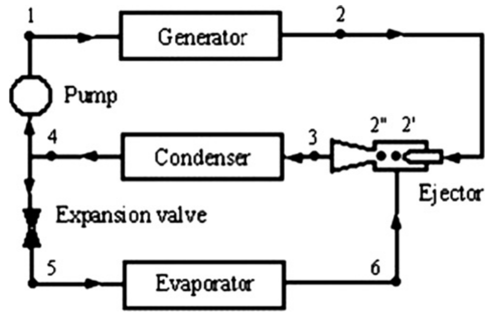

Fig. 15. Schematic diagram of the transcritical ejector refrigeration system [89].

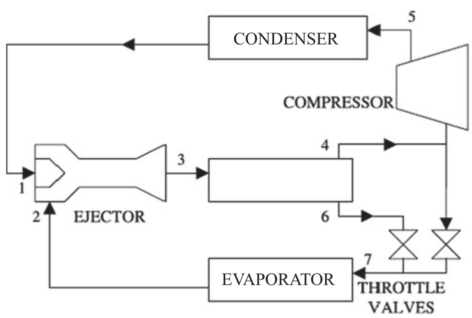

Fig. 16. Schematic diagram of the new ejector expansion refrigeration cycle [90].

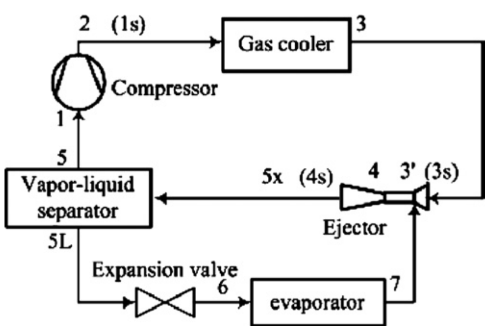

Fig. 17. Schematic diagram of the ejector expansion system [91].

vapour backflow valve to relax the constraints between the entrainment ratio of the ejector and the quality of the ejector outlet stream. The effect of different operating conditions on the relative performance of the ejector expansion transcritical CO_2 cycle was also investigated using assumed values for the entrainment ratio and pressure drop in the receiving section of the ejector. The results demonstrated that the ejector expansion cycle improved the COP by more than 16% compared to the basic cycle for typical air conditioning applications.

Deng et al. [91] presented a theoretical analysis of a transcritical CO_2 ejector expansion refrigeration cycle (shown in Fig. 17), which uses an ejector as the main expansion device instead of an expansion valve. The results indicated that the ejector entrainment ratio significantly influenced the refrigeration effect with an optimum ratio giving the ideal system performance. It was found that for the working conditions described in their paper, the ejector improved the maximum COP to 18.6% compared to the internal heat exchanger system and 22% compared to the conventional system.

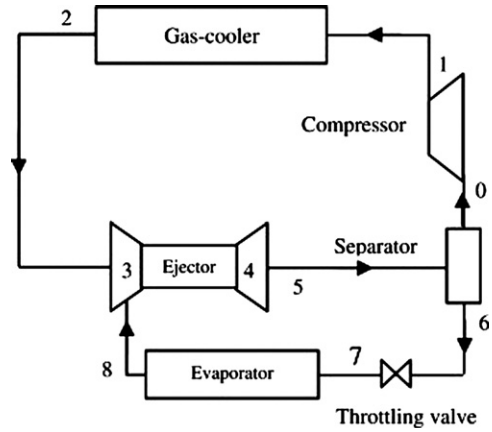

Fig. 18. Schematic diagram of transcritical carbon dioxide cycle with ejector [94].

Yari and Sirousazar [94] investigated a new transcritical CO_2 refrigeration cycle (TRCC) with an ejector, internal heat exchanger and intercooler (shown in Fig. 18). This cycle utilized the internal heat exchanger and intercooler to enhance its performance significantly. It was found that, the new ejector expansion TRCC improved the maximum COP and second law efficiency up to 26% compared to conventional ejector-expansion TRCC.

5.4. Solar-driven ejector refrigeration system

Because of the ability of harnessing solar energy, the solar-driven ERS is less energy demanding and more environmental friendly in comparison with conventional vapour compression refrigeration system. However, due to the intermittent feature of solar energy, the unstable heat gains from solar sources inherently affect the operation of solar-driven ERS. Thus, thermal storage system integrated with solar-driven ERS is becoming a hot research topic.

5.4.1. Conventional solar-driven ejector refrigeration system

Conventional solar-driven ERS has been widely studied during past decade. Heat from the solar collector is carried by the intermediate medium and transferred to the refrigerant by the heat exchanger. The heat transfer mediums should have the boiling point higher than the possible temperature in the system, low viscosity and good heat transfer properties. Water with a corrosion inhibitor additive and transforming oil are recommended for operating temperature below and above 100 °C, respectively. However, since water will freeze below 0 °C, Varga et al. [43] found that the system working at very low evaporator temperature was not suitable for using water as refrigerant.

Theoretical analysis of a solar-driven ERS in the Mediterranean was carried out by Varga et al. [43]. Based on a simplified 1-D model, the authors studied both the refrigeration and solar collector cycles for a 5 kW cooling capacity. The results indicated that, in order to achieve acceptable COP, generator temperature should not fall below 90 °C and solar collector output temperature of about 100 °C would be required. For higher condenser temperatures ($>$35 °C) and lower evaporator temperature ($<$10 °C), the solar collector area required for 5 kW cooling load was larger than 50 m^2.

R134a was proposed as a refrigerant for a solar-driven ejector system by Alexis and Karayiannis et al. [22]. It was found that COP of ejector cooling system varied from 0.035 to 0.199 for generator temperatures ranging from 82 to 92 °C, condenser temperatures

ranging from 32 to 40 °C and evaporator temperatures ranging from −10 to 0 °C.

Ersoy et al. [17] conducted a numerical investigation on the performance of a solar-driven ejector cooling system using R114 under Turkish climatic conditions. When generator, condenser, and evaporator temperatures were taken at 85 °C, 30 °C and 12 °C, respectively, the maximum overall COP and the cooling capacity obtained were as 0.197 and 178.26 W/m^2.

Pridasawas and Lundqvist [39] carried out a year-round simulation of a solar-driven ejector system with 3.5 kW cooling capacity at a generator temperature of 70–120 °C and an evaporator temperature of 15 °C with R600a as refrigerant. With an evacuated collector area 50 m^2 and a hot storage tank volume of 2 m^3, the average yearly system thermal ratio (STR) was about 0.22, the COP of the ejector system was about 0.48, and the solar collector efficiency was about 0.47. The authors concluded that an efficient ejector system could only work in a region with decent solar radiation and where a sufficiently low condenser temperature could be maintained.

Experimental data and simulation results for recent solar-driven ERS researches are listed in Table 5.

Table 5
Operating conditions and system performances of various solar-driven ejector systems.

Reference	Area	Solar radiation (w/m^2)	Cooling capacity (kW)	Working fluid	T_e (°C)	T_c (°C)	T_b (°C)	Solar collector area (m^2)	COP_{eje}
Varga et al. [43]	Mediterranean countries	800	5	water	5:15	30:35	90:110	50	0.1:0.55
Alexis and Karayiannis [22]	Athens	536:838	–	R134a	−10:0	32:40	82:92	–	0.035:0.199
Ersoy et al. [17]	Turkey	178.3	3.2	R123	12	30	85	21	0.197
Pridasawas and Lundqvist [39]	Bangkok	600:900	3.5	R600a	5:15	5 °C above ambient	70:120	50	0.35:0.48
Vidal et al. [24]	Brazil	–	10.5	R141b	8	32	80	80	0.39
Huang et al. [34]	–	700	10.5	R141b	8	32	90	68	0.5

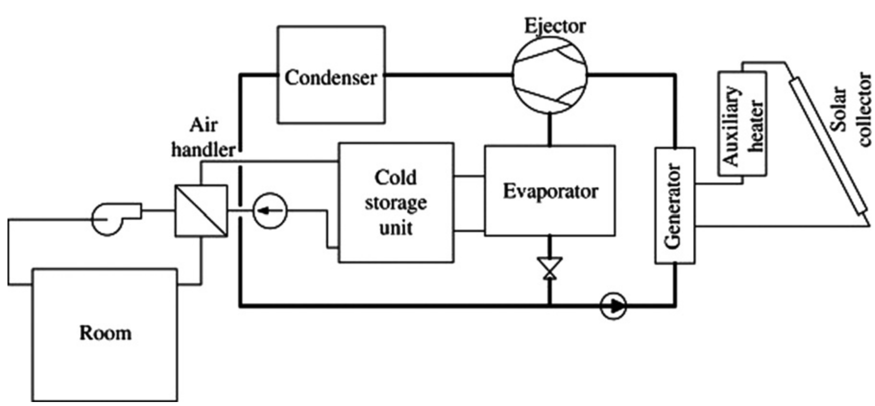

Fig. 19. Schematic diagram of solar-assisted ejector refrigerator with cold storage [97].

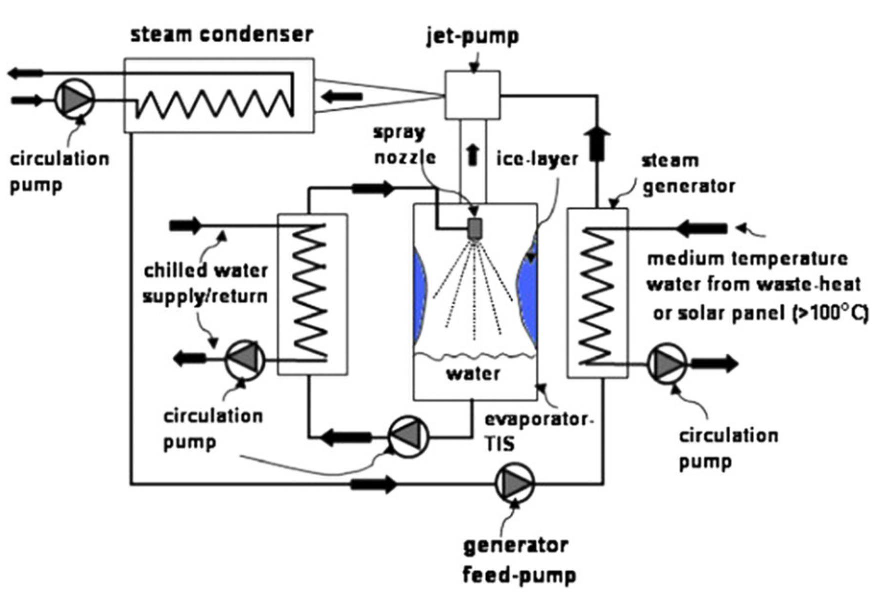

Fig. 20. Schematic diagram of ejector refrigeration system with thermal ice storage [100].

5.4.2. Solar-driven ejector refrigeration system with thermal storage system

During some adverse weather conditions, the cooling capacity provided by available solar energy cannot be essentially matched with the cooling demand. Taken this into consideration, energy storage technology was applied in solar-driven ERS. Two kinds of thermal storage are considered: hot storage-high temperature energy from the solar collectors, and cold storage-low temperature energy from the evaporator.

Guo and Shen [95] numerically investigated a solar-driven air conditioning system with hot storage for office buildings. With generator temperature of 85 °C, evaporator temperature of 8 °C and condenser temperature varying with ambient temperature, the average COP and the average solar fraction of the system were 0.48 and 0.82, respectively. It was concluded that the system could save approximately 75% of the electricity used for conventional air conditioning under Shanghai's climatic conditions.

In contrast, Pridasawas and Lundqvist [39] reported that the size of the hot storage tank did not improve significantly the performance of the system. Hence, cold storage, with the help of phase changing materials, cold water or ice storage, was recommended by Bejan et al. [96]. Moreover, using computer simulations, Diaconu et al. [97] analysed a solar-assisted ejector cooling system with cold storage (as shown in Fig. 19) over one year in Algeria. Compared to that without cold storage, the annual energy removal of the system with cold storage achieved higher values. In order to provide better compliance with varying ambient conditions, a variable geometry ejector with cold storage was investigated by Dennis et al. [98]. The annual cooling simulation results concluded that a variable geometry ejector was able to increase yield by 8–13% compared to a fixed geometry ejector. The modelling further showed that the solar collector area may be decreased if a cold storage was used.

Worall et al. [99] and Eames et al. [100] carried an experimental investigation of a novel ejector refrigeration cycle with thermal ice storage system (as shown in Fig. 20). Ice was formed in the evaporator vessel under normal operation and acted as a coolth storage medium. The low evaporator temperature resulted in a relatively low COP of 0.162 during experiments. The authors argued that such system powered by solar energy would help to store the coolth to level out the off-peak conditions.

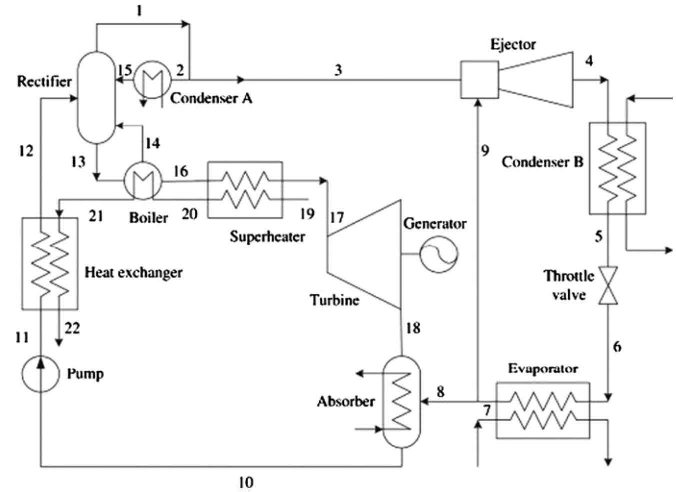

Fig. 22. Schematic diagram of the combined power and ejector-absorption refrigeration cycle [47].

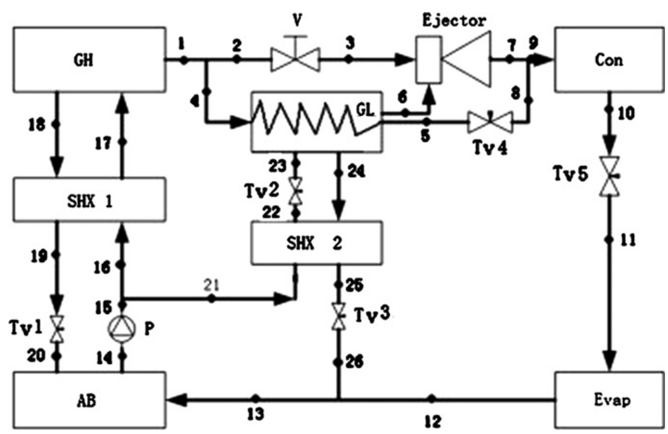

Fig. 23. Schematic diagram of a novel ejector-absorption combined refrigeration cycle [102] (AB, absorber; Con, condenser; Evap, evaporator; GH, high-pressure generator; GL, low-pressure generator; P, pump; SHX, heat exchanger; V, valve).

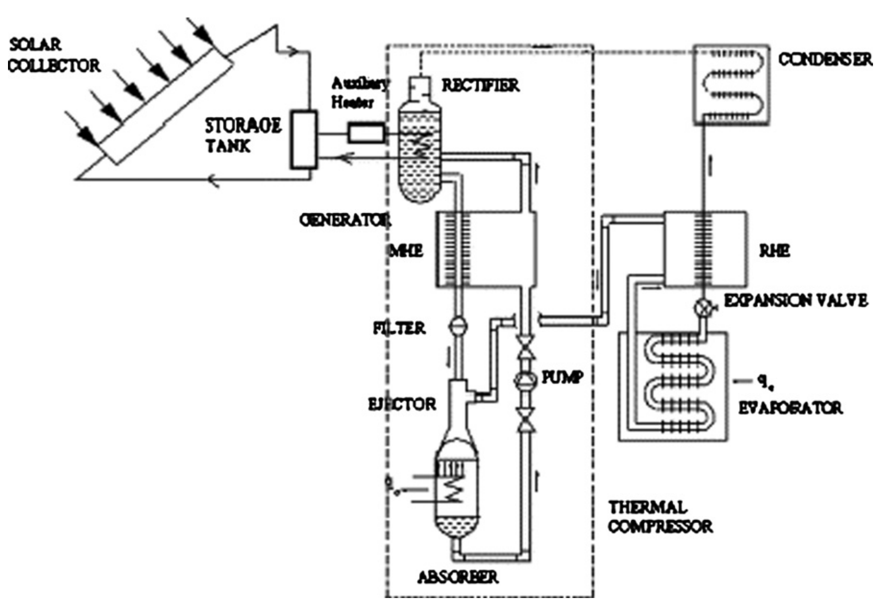

Fig. 21. Schematic diagram of a novel ejector-absorption combined refrigeration cycle [46].

6. Application of ejector refrigeration system combined with other systems

6.1. Combined ejector-absorption refrigeration system

Absorption system can also make use of low-grade heat sources, such as solar energy, waste or exhaust heat. However, because of its complex configuration and low COP, it is less competitive than the conventional vapour compression system. Applying ejector to the conventional absorption systems is one of the remarkable alternatives. The appropriate installation configuration can help to improve the system performance almost similar to multi-effect absorption cycle machine. Moreover, due to the simplicity of the combined ejector-absorption refrigeration machine, its capital investment cost is comparatively low compared to other conventional high performance absorption cycle systems.

Recently, Sozen et al. [46] proposed a solar-driven ejector-absorption system (shown in Fig. 21) operated with aqua-ammonia under the climatic condition of Turkey. Ejector was located at the absorber inlet, which helped the pressure recovery from the evaporator. According to results obtained in this study, using the ejector, the COP was improved by about 20%. For 8–9 months (March–October) of the year, the collector surface area of 4 m^2 was sufficient for different applications of refrigeration all over Turkey.

Wang et al. [47] presented a combined power and ejector-absorption refrigeration cycle with aqua-ammonia as working fluids. This system (shown in Fig. 22) combined the Rankine cycle with ejector-absorption refrigeration cycle, and could produce both power output and refrigeration output simultaneously. This combined cycle introduces an ejector between the rectifier and the condenser, and provides a performance improvement without greatly increasing the complexity of the system. The comparisons of the parametric results between a similar combined system without ejector [101] and this system showed that refrigeration output increased from 149 kW to 250 kW at evaporator temperature of -8 °C and generator temperature of 87 °C.

In order to make sufficient use of high-grade heat with a simple structure refrigeration system, Hong et al. [102] proposed a novel ejector-absorption combined refrigeration cycle (shown in Fig. 23). When the temperature of the heat source is high enough, the cycle would work as a double-effect cycle. Two generators were used in the cycle, so that the pressure of the high-pressure generator and that of the low-pressure generator could be optimized to get maximum COP at any given working condition. The simulation results showed that system COP was 30% higher than that of the conventional single-effect absorption refrigeration cycle. However, no experimental validation was available.

Theoretical and experimental study of solar-ejector absorption refrigeration system (shown in Fig. 24) was conducted by Abdulateef et al. [103]. The effects of the operating conditions on the COP and the cooling capacity of the system were investigated. A mathematical model was developed for design and performance evaluation of the ERS. A wide range of compression, expansion and entrainment ratios, especially those used in industrial applications were covered in the mathematical model.

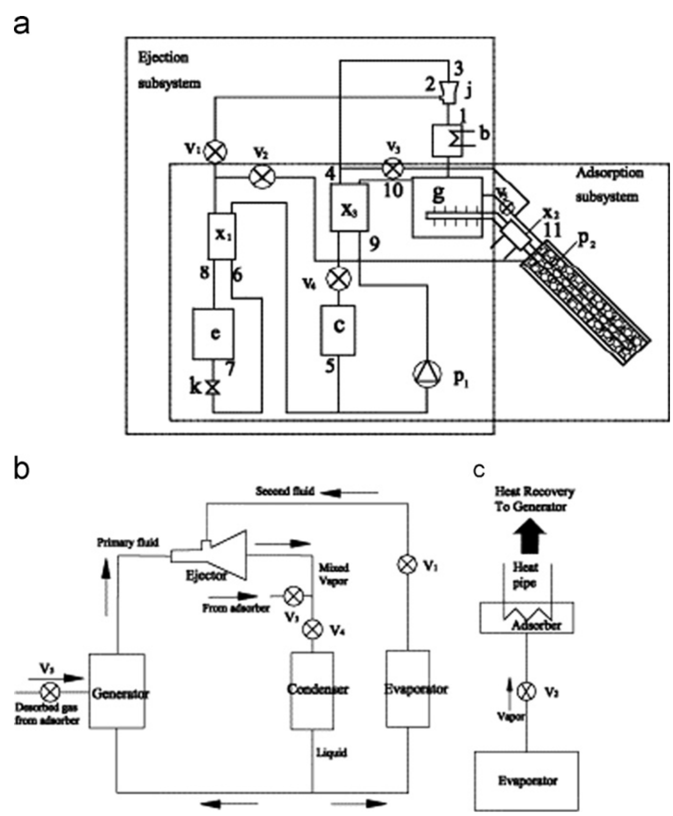

Fig. 25. Schematic diagram of a solar-powered adsorption–ejection refrigeration system [104]. (a) System layout, (b) ejector refrigeration system during daytime and (c) adsorption refrigeration system at night (a, adsorber; b, auxiliary heater; c, condenser; e, evaporator; g, generator; j, ejector; k, expansion valve; p1, pump; p2, heat pipe; x, heat exchangers; v, valves).

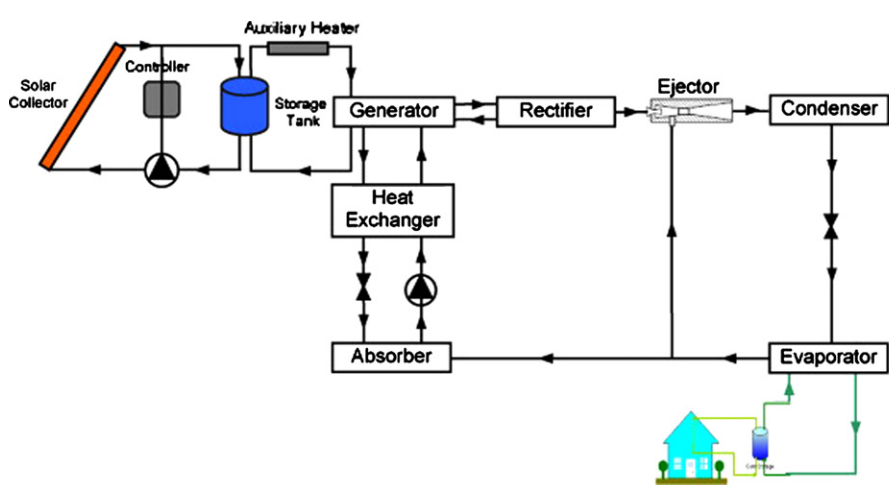

Fig. 24. Schematic diagram of solar-driven ejector absorption refrigeration system [103].

With the aim of overcoming the intermittency of adsorption refrigeration, Li et al. [104] presented a novel combined cycle solar-powered adsorption–ejection refrigeration system using Zeolite 13X-water as the pair. The cycle consisted of two subsystems (as shown in Fig. 25): ejector sub-system to provide refrigeration during the day and an adsorption sub-system which refrigerates at night-time. Detailed system description can be found in [104]. It was demonstrated that the COP of the ejection sub-system improved when the temperature of the adsorbent increased or when the pressure decreased. A COP of 0.4 was achieved with 9 °C evaporating temperature, 40 °C condensing temperature, 120 °C regenerating temperature and 200 °C desorbing temperature. It was further concluded that by increasing the temperature or reducing the pressure within the adsorbent bed, the COPs of the ejection sub-system could be improved slightly.

6.2. Ejector refrigeration system with compressor or vapour compression system

Since the system performance of ERS is determined by ejector entrainment ratio λ and operating conditions, one way to enhance the performance is to increase the secondary flow pressure. In 1990, Sokolov and Hearshgal [105], introduced new configurations of efficient uses of the mechanical power in order to enhance the secondary pressure without disturbing the refrigeration temperature, which are: (1) the booster assisted ejector cycle and (2) the hybrid vapour compression-jet cycle. Their simulated results showed that the compression enhanced ejector could significantly improve system performance.

Recently, Sokolov et al. [106] improved their system by using a booster and intercooler in a solar-powered ERS (as shown in Fig. 26). The system consisted of a conventional compression and ejector sub-cycles with an intercooler as an interface between them. The intercooler is a heat and mass exchanger through which the two sub-cycles interact. Heat absorbed in the evaporator is boosted up to the intercooler pressure and temperature by the compression cycle. The elevated suction pressure from the intercooler to the ejector results in a higher mass flow rate with which the ejector operates. The ejector sub-cycle further raises the heat from the evaporator to the condenser's pressure and temperature. The system operated at 4 °C evaporator temperature and 50 °C condenser temperature, with cooling capacity of 3.5 kW. The overall system COP could reach up to 0.5. The group later revised their work [107] by substituting the refrigerant of R114 with R142b. The results indicated that R142b provided higher efficiency than the one operating with R114.

A similar system configuration was presented by Hernandez et al. [23] with R134a and R142b as working fluids. The theoretical analysis demonstrated that the optimum COP of 0.48 could be achieved at condenser temperature of 30 °C and generator temperature of 85 °C, with R134a as working fluid. The authors further indicated that when higher condenser temperature was imposed, the system with R142b would perform better.

Vidal et al. [24] implemented a computer simulation on a solar assisted combined ejector-compression system. The mechanical compression cycle and the thermal driven ejector cycle were performed with two different refrigerants, R134a and R141b respectively. The final optimum results showed that an intercooler temperature of 19 °C resulting in a solar fraction of the system of 82% and a COP of the combined ejector cycle of 0.89. A 10.5 kW cooling capacity was achieved with the flat plate collector area of 105 m^2.

Zhu et al. [38] proposed a hybrid vapour compression refrigeration system which combined with an ejector cooling cycle (as shown in Fig. 27). The ejector cooling cycle was driven by waste heat from the condenser in the vapour compression refrigeration cycle. The additional cooling capacity from the ejector cycle in directly input to the evaporator of the vapour compression refrigeration cycle. Simulation results showed that

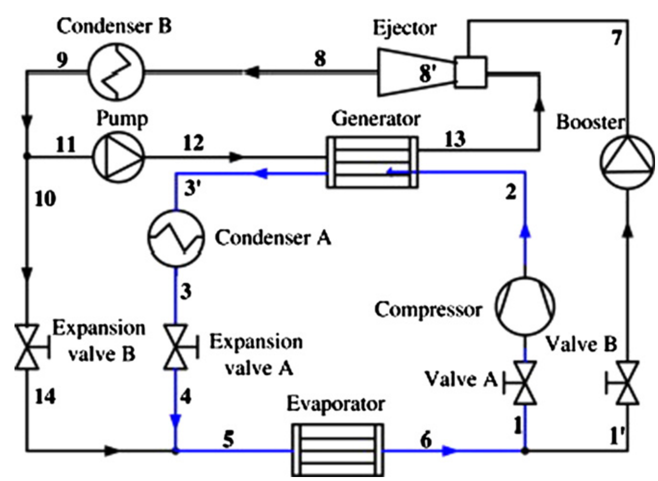

Fig. 27. Schematic diagram of a refrigeration system with the integrated ejector [38].

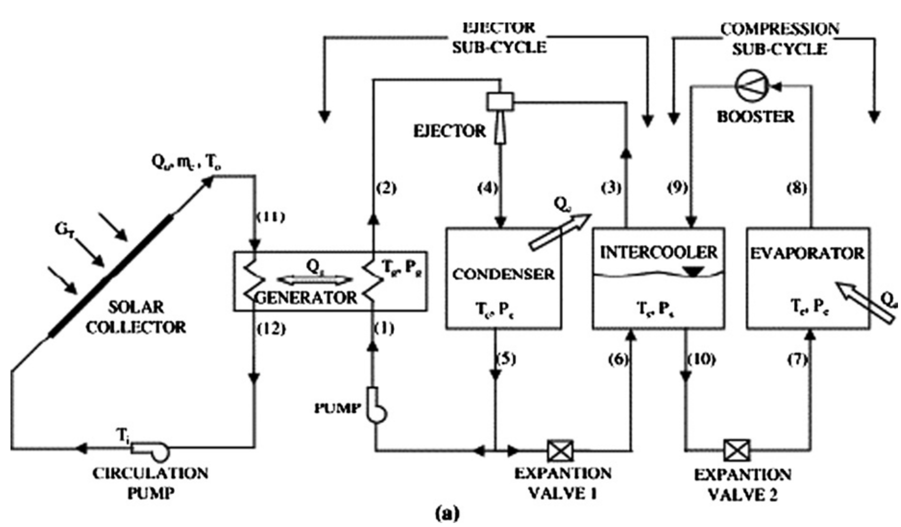

Fig. 26. Schematic diagram of a solar-powered ERS [106].

COP increased by 5.5% with R152a and 8.8% with R22 compared with the basic system. However, no experimental results were available to validate these.

Worall et al. [93,108] proposed a similar hybrid CO_2 ejector and vapour compression system as shown in Fig. 28. The ejector refrigeration system was proposed to extract heat from the exhaust of an independent diesel engine and sub-cool the CO_2 vapour compression system. The modelling results showed that at an evaporator temperature of $-15\,^{\circ}C$, an ambient temperature of $35\,^{\circ}C$ and a generator temperature of $120\,^{\circ}C$, COP could be increased from 1.0 to 2.27 as sub-cooling increased from 0 to $20\,^{\circ}C$. At the same time, the compressor work could be reduced by 24% at $20\,^{\circ}C$ sub-cooling. The group [109] later carried out preliminary experimental investigations on the ejector cycle.

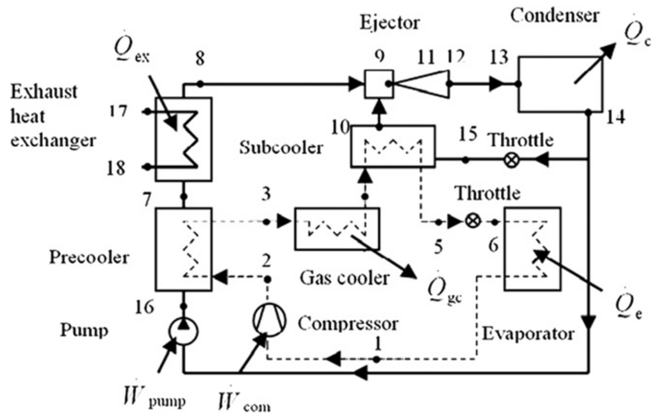

Fig. 28. Schematic diagram of hybrid CO_2 ejector and vapour compression system [108].

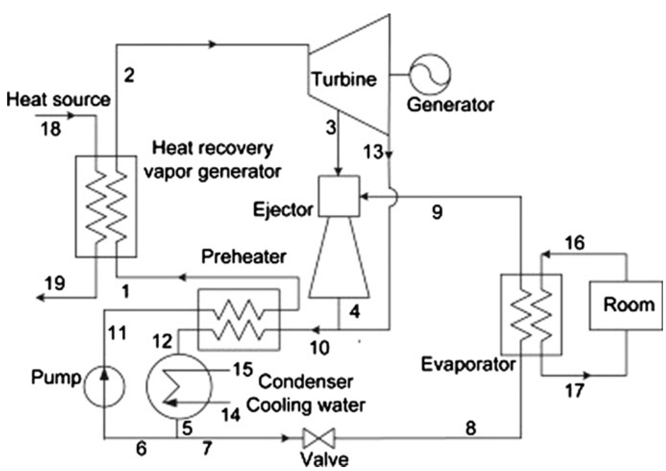

Fig. 29. Schematic diagram of combined power and ERS [111].

6.3. Combined power and ejector refrigeration system

Recently, many combined power and refrigeration cycles have been proposed to make better use of low grade heat sources. Zhang and Lior [110] discussed the combined power and refrigeration cycles with both parallel and series-connected configurations. The cycle has large refrigeration capacity. However, it operated at temperatures about $450\,^{\circ}C$, which is incompatible with low temperature heat sources such as solar thermal and waste heat.

Wang et al. [111] proposed a combined power and refrigeration cycle as shown in Fig. 29, which combined the Rankine cycle with the ERS by adding an extraction turbine between heat recovery vapour generator (HRVG) and ejector. This combined cycle could produce both power output and refrigeration output simultaneously. The HRVG is a device in which high pressure and temperature vapour is generated by absorbing heat from sources such as solar thermal, geothermal and waste heat. The parametric analysis results concluded that the amounts of exergy destruction in the HRVG, ejector and turbine accounted for a large percentage. The author suggested several methods to improve system

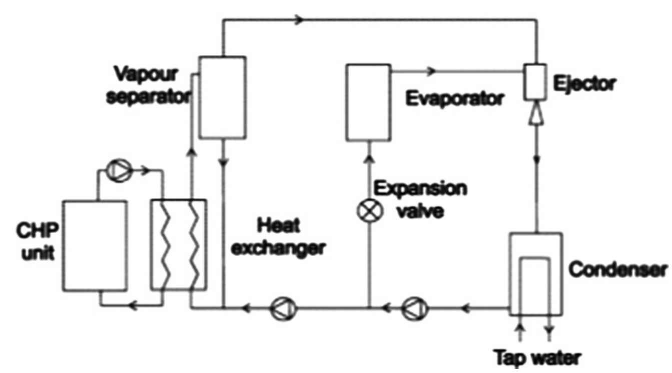

Fig. 31. Schematic diagram of a CHP-ejector system [114].

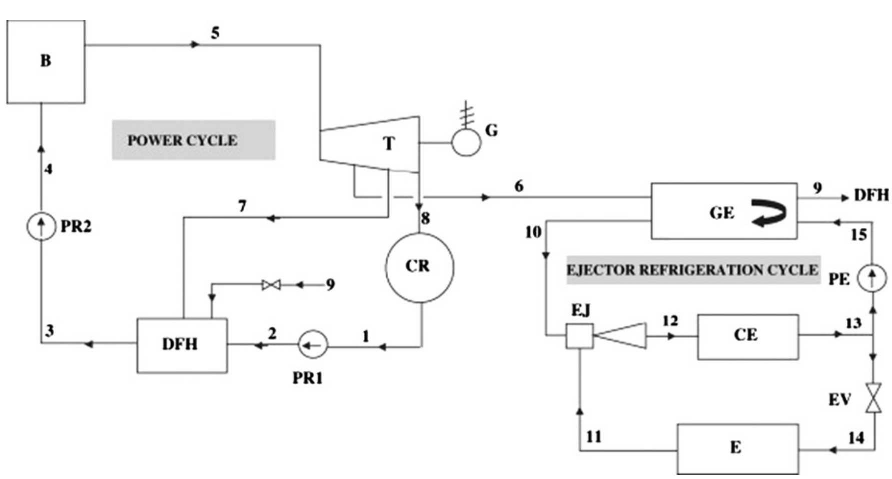

Fig. 30. Schematic view of combined refrigeration and electrical power cogeneration system [112] (B, boiler; T, turbine; G, generator; CR, main condenser; PR1, condensate pump; DFH, deaerating feed water heater, PR2; feed water pump; GE, heat generator; EJ, ejector; CE, condenser; E, evaporator; PE, pump; EV, expansion valve).

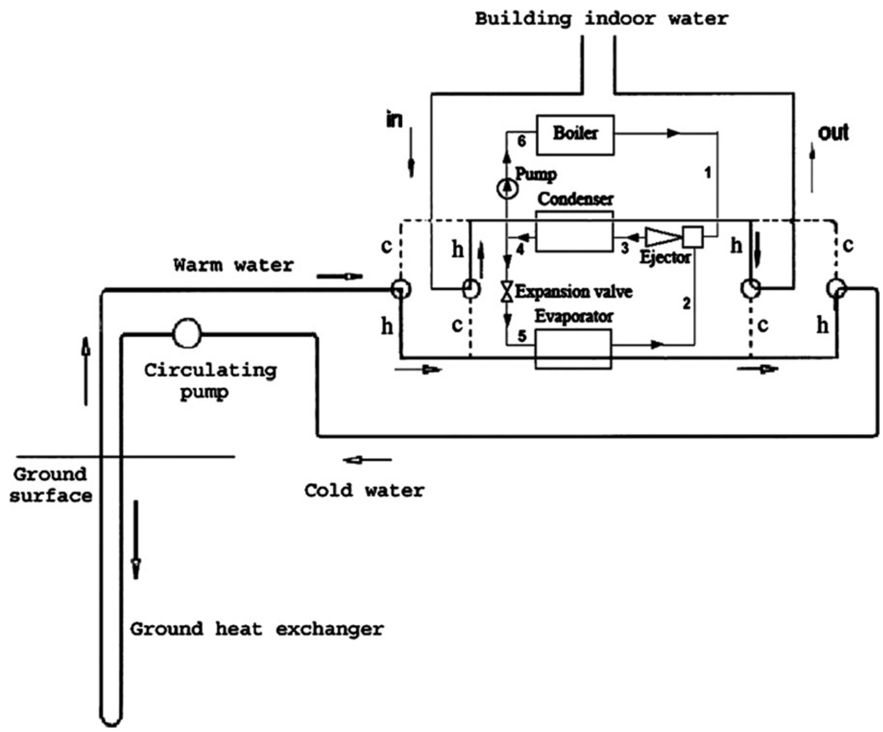

Fig. 32. Schematic diagram of ground coupled steam ejector heat pump system [115].

efficiency including increasing the area of heat transfer and the coefficient of heat transfer in the HRVG, optimization design parameters in the ejector and turbine.

Similarly, Alexis et al. [112] studied a combined power and ejector cooling cycle (Fig. 30) in which extracted steam from the turbine in Rankine power cycle was used to heat the working fluid in an independent steam ejector refrigeration cycle. Rankine cycle and steam ejector refrigeration cycle produced electrical power and refrigeration capacity, respectively. Computer modelling results showed that when the ratio between electrical power and heat transfer rate was varied between 0.1 and 0.4, the ratio between electrical power and refrigeration capacity was varied between 0.23 and 0.92.

A combined power and ERS with R245fa as working fluid was presented by Zheng et al. [113]. Simulation results showed that a thermal efficiency of 34.1%, an effective efficiency of 18.7% and an exergy efficiency of 56.8% could be obtained at a generator temperature of 122 °C, a condensing temperature of 25 °C and an evaporating temperature of 7 °C. It was also noted that while the generator temperature increased the fluid inlet pressure of the ejector increased.

Exergy analysis of combined power and ejector refrigeration cycle presented by Wang [101] showed that the largest exergy destruction occurred in the heat recovery vapor generator (HRVG) followed by the ejector and turbine. In order to recover some of the thermal energy from the turbine exhaust, Khaliq [35] combined a Libr-H_2O absorption system with power and ejector refrigeration system using R141b as refrigerant. The results of first and second law investigation showed that the proposed congeneration cycle yielded better thermal and exergy efficiencies than the cycle without absorption system. However, no experimental results were available.

Godefroy et al. [114] designed a small CHP-ejector trigeneration system which combined heat and power (CHP) to drive an ejector cooling cycle. In the system (shown in Fig. 31) consisted of a CHP unit and an ejector cooling cycle. The ejector cooling cycle was driven by heat from the CHP unit supplied through a flat-plate heat exchanger to bring the refrigerant to its vapour state. The design had been tested and validated by a model based on the real fluid properties. The results showed that this system offered an overall efficiency around 50% and would have an almost neutral effect on overall emissions.

6.4. Ground coupled steam ejector heat pump

Ground coupled heat pump (GCHP) is being used for heating and cooling residential and commercial buildings by exchanging heat with the ground as the thermal source or sink. However the initial investment is higher than that for the air source heat pumps due to the costs of ground loop pipes, wells, channels and circulation pumps. Ejector systems, with its advantage of long operating lifetime, high reliability, low maintenance cost, are one alternative to reduce the initial cost of GCHP. Sanaye et al. [115] investigated a GCHP (shown in Fig. 32) which included two main sections of closed vertical ground heat exchanger and steam ejector heat pump. Thermal and economic simulation and optimization of the system, optimum design of ejector main cross section and investigation of the effects of weather, soil type, and system capacity on system performance were carried out in this research. Simulation results were validated with experimental data from the literature. The authors concluded that the system had the smallest mean total annual cost value and maximum overall COP in temperature climates in comparison with cold and tropical climates.

7. Conclusions

Studies in ejector refrigeration systems that have been carried out over the past decade involved system modelling, design fundamentals, refrigerants selection and system optimization. The research and development was broad based and productive, concentrating on performance enhancement methodology and feasibility of combining ERS with other systems. This paper

presents not only a basic background and principles for ejector design, but also the recent improvement in ejector refrigeration technologies.

The following conclusions can be drawn from the reviewed works that have been carried out in ejector refrigeration system:

- With the phasing out of CFCs, HCFCs and HFCs, halocarbon and hydrocarbon compounds with low GWP and ODP were considered as new alternatives, and were widely studied and compared as working fluids in different ERS. Among these, R143a and R142b were proved to yield better performance than other working fluids. As the most environmentally friendly and economically available refrigerant, water has been widely used and tested as refrigerant for ejector refrigeration system.
- Some attempts have been made on the investigations of proper mathematical models that may help to optimize design parameters. Taking into consideration of friction losses and irreversibilites, some researchers have carried out computer simulations on the improvement of constant-area model and constant-pressure model. A number of researchers have concentrated on the studies of two-phase flow and specific characteristics of working fluids. CFD has been identified as a suitable tool for the turbulence models of the mixing process which can better simulate and optimize the geometry of ejector. Although these simulated results were claimed to become more accurate than others, very few of them were experimentally verified and approved.
- Different configurations of ejectors with various geometric features were proposed and tested numerically and experimentally. Area ratio and nozzle exit position were the most widely investigated parameters. It can be concluded that the optimal area ratio and NXP have varied for the different operating conditions. A spindle, which can adjust primary nozzle position, could be implemented to provide both flexible area ratio and NXP.
- Since the ejector refrigeration systems suffer from relatively low COP, a number of studies have focused on system performance enhancement. Operation of ERS without a pump has been declared to considerably reduce the mechanical energy consumption. In contrast, ERS with an additional pump could help to increase the entrainment ratio and COP. In order to cope with variations of working conditions, multi-components ERS are parametrically studied. On the other hand, transcritical ERS is proposed, which provides higher potential in utilizing low-grade heat. The remarkable COP improvements from combined ejector and other types of refrigeration systems (vapour compression, absorption system, etc.) are reported by many research groups. However, most of those studied are limited to numerical analysis, with few experimental results available.
- With the concept of energy conservation and environment protection, the utilization of low grade energy, especially solar energy with ERS has been widely studied during past decade. The major technical problem of solar-driven ejector refrigeration system is that the system is strongly reliant on ambient conditions, like the solar radiation, air temperature, cooling water temperature, wind speed and other transient factors. Thus the combination of energy storage in the solar-driven ERS remains to be the research topic in this field of technology.

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